MAR 21 1912
PUBLIC LIBRARY
CITY OF DETROIT

POWER

Volume 35, No. 12.
Issued Every Tuesday.
Hill Publishing Company.

NEW YORK, MARCH 19, 1912.

Price, 5 cents.
Outline of Topics, 54.
Index to Advertisers, 44.

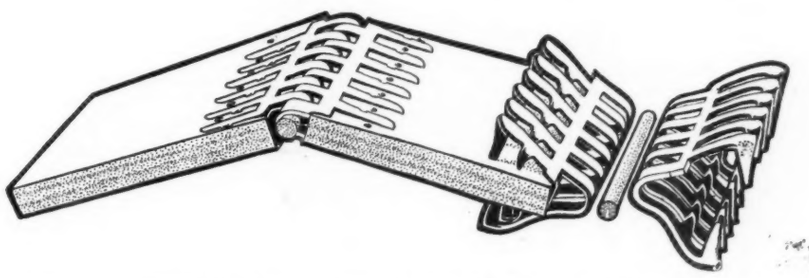
ALLIGATOR

(TRADE MARK)

STEEL BELT LACING

Has A Grip No Belt Strain Can Break

Repeatedly when tested under strains in excess of those of actual service it has shown remarkable results. Made in 8 different sizes and adapted for use on any kind, width or thickness of belting. It has established an enviable record working perfectly under conditions where other fasteners could not be used. Every time "Alligator Steel Lacing" grips the belt, it gets a *double* hold—note the cut below and you will see how. The prongs are so arranged that they clinch on the opposite side of the belt. The prongs enter the belt so that the pull is against edge of metal preserving greatest strength of belt. Because of the perfectly flexible joint, hinging on rawhide or Steel "Rocker" pin there is never any unequal or excessive strain on any part of the lacing. Can be used with an idler or tightener. Remember too, no tool but a hammer is needed to apply "Alligator Steel Lacing."



Don't wait till the break comes—send for that Free Sample now and be prepared.
When you write, indicate both kind and size of your belting.

Flexible Steel Lacing Company

542 West Jackson Boulevard Chicago, Illinois

EUROPEAN AGENTS—J. B. Stone & Co., 135 Finsbury Pavement, London, Eng.

To Be On The Safe Side—Try The Dean

"Where we supposed our tubes were comparatively clean we are drawing out scale by the wheelbarrow load."

"After the first trial it opened our eyes to the quantity of scale, etc., that may be on boilers without knowing it."

"It is a surprise to us that there could be such an accumulation of scale and dirt with all the care we used in cleaning boilers heretofore in the old way."

"We removed over 300 pounds of scale from a boiler we thought clean."

"We took out of our boilers ten wheelbarrows (of scale) in all and the boilers were considered by the inspectors to be in good shape."

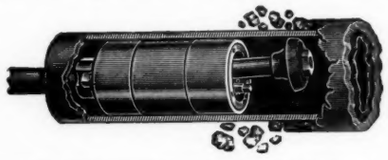
"I removed 260 lbs.of scale from our No.2 boiler directly after an Inspector's report of'Clean Boiler'."

The above are unsolicited statements from concerns who tried the *Dean Boiler Tube Cleaner* in their supposedly clean boilers. They tried it because they wanted to be on the safe side—didn't want to stand the chance of paying for scale. It's better to try a Dean and make sure you have no scale than not to try it and regret later that you never tried it.

The Dean removing scale from the tube of a water tube boiler.

Free Trial Offer

We'll loan you a Dean for free trial in one boiler. We won't hurry you with the test, force the cleaner on you, nor get sore if we don't effect a sale. We simply want you to see what a money saving proposition we have. Then it's up to you to decide what to do with it.



The Dean removing scale from the tube of a return tubular boiler.

The William B. Pierce Company

Jewett Building **Buffalo, N. Y.**

Chicago Office: 801 Steinway Building

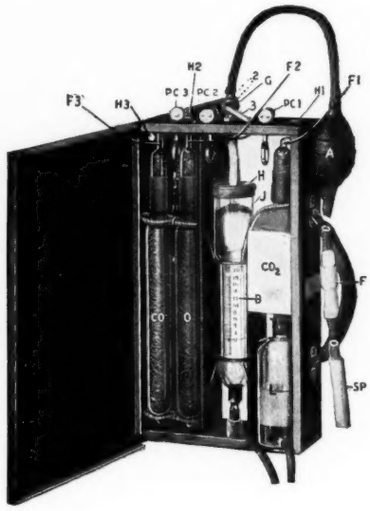
In one hour's time you can submit all of the following questions to

The Hays Improved Gas Analysis Instrument

and obtain a *positive* and reliable answer to each of them:

What is the efficiency of your boiler furnace?
What draft will carry the load and burn the least coal?
How much air is leaking through the setting?
Are you using the coal best adapted to your conditions?
Should this coal be fired dry or wet for greatest economy?
Is the fuel bed carried too thick or too thin?
Are you using the grate best adapted to your coal?
Is the grate surface just right for highest economy?
How much and what kind of combustible is passing up your chimney?
What must be done to burn the escaping combustible gases?
Can you get complete combustion and satisfy the smoke inspector without putting in an expensive special furnace?
Is the low evaporation of which you complain due to the boiler, the furnace, the coal or the fireman?
How much coal is your poor fireman wasting and how much coal can your best fireman save?

A word from you brings our circular describing this Instrument and our easy-payment plan.



The Wm. B. Pierce Company

Jewett Building :: :: :: **Buffalo, N. Y.**

KEYSTONE GREASE

"There's The Solution"

Engineer Brown was "stumped" on a problem of lubrication. He called on his neighbor, Engineer Jones, who was "stumped" too—but only for a minute. For he had a wise friend at his right hand—a bound volume of

The Keystone Bulletin

He turned to a certain volume and page—and there was the exact situation all straightened out.

Mind you, we do not claim that the Keystone Bulletin is a "cure all," that every time you are "stumped," you will find the exact solution of your problem in some volume.

But we do claim that we are getting out a series of Bulletins that contains *many* suggestions of value, *much* information that covers a *wide* range of usefulness—Bulletins that *in all probability* will contain somewhere or other, the information that will erase the wrinkles from the "stumped" engineer's brow, and bring a smile of solid satisfaction instead.

The Keystone Bulletin appears at least once every month. It's absolutely free.

All that you need do to receive it regularly, is to send your name and address and we will send current number, back number, and loose leaf binder at once, and future copies as they appear.

Isn't this an offer which you cannot afford to let slip?

KEYSTONE
TRADE MARK
REGISTERED U.S. PAT. OFF.

KEYSTONE LUBRICATING CO.

Executive Offices and Works:

21st, Clearfield and Lippincott Streets Philadelphia, Pa.

Branch offices and warehouses in the principal cities of the United States.
Agencies in the principal countries throughout the world.

NO CONNECTION WITH THE OIL TRUST

This Coupon Brings The Keystone Bulletin **Fill out and return it—Today**

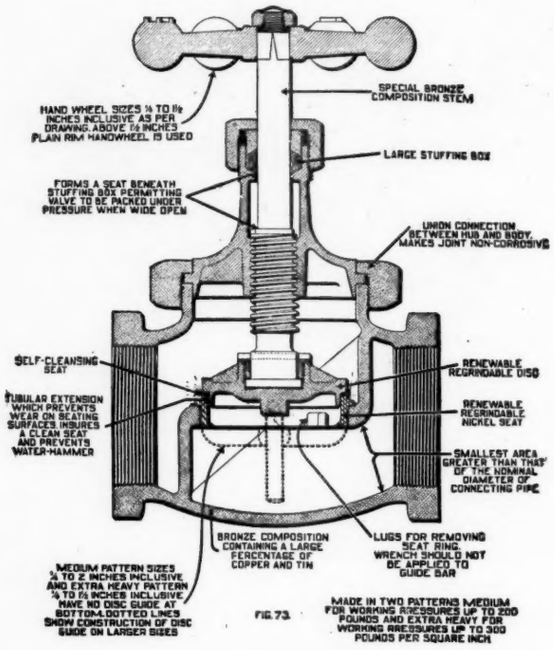
THE KEYSTONE LUBRICATING COMPANY

Send me Keystone Bulletins for 1911, with Loose Leaf Binder; also other issues as they appear.

Name.....Address.....

Firm Name.....Address.....

3-19-12


HAND WHEEL SIZES ¼ TO 1½ INCHES INCLUSIVE AS PER DRAWING. ABOVE 1½ INCHES PLAIN RIM HANDWHEEL IS USED
SPECIAL BRONZE COMPOSITION STEM
LARGE STUFFING BOX
FORMS A SEAT BENEATH STUFFING BOX PERMITTING VALVE TO BE PACKED UNDER PRESSURE WHEN WIDE OPEN
UNION CONNECTION BETWEEN HUB AND BODY MAKES JOINT NON-CORROSIVE
SELF-CLEANSING SEAT
TUBULAR EXTENSION WHICH PREVENTS WEAR ON SEATING SURFACES INSURES A CLEAN SEAT AND PREVENTS WATER-HAMMER
RENEWABLE REGRINDABLE DISC
RENEWABLE REGRINDABLE NICKEL SEAT
SMALLEST AREA GREATER THAN THAT OF THE NOMINAL DIAMETER OF CONNECTING PIPE
BRONZE COMPOSITION CONTAINING A LARGE PERCENTAGE OF COPPER AND TIN
LUGS FOR REMOVING SEAT RING. WRENCH SHOULD NOT BE APPLIED TO GUIDE BAR
MEDIUM PATTERN SIZES ¼ TO 2 INCHES INCLUSIVE AND EXTRA HEAVY PATTERN ¼ TO 1½ INCHES INCLUSIVE HAVE NO DISC GUIDE AT BOTTOM. DOTTED LINES SHOW CONSTRUCTION OF DISC GUIDE ON LARGER SIZES
FIG. 73.
MADE IN TWO PATTERNS MEDIUM FOR WORKING PRESSURES UP TO 200 POUNDS AND EXTRA HEAVY FOR WORKING PRESSURES UP TO 300 POUNDS PER SQUARE INCH

A valve that has sufficient strength; made of materials best suited for steam use; faultless workmanship; large areas, insuring a free and unobstructed passage of steam, and so designed that all parts can easily be renewed,—must be conceded as one that is perfect, or as nearly so as can be attained.

This being conceded, then the

LUNKENHEIMER "RENEWO" VALVE

can justly be called a valve of perfection, as it embraces all of the above necessary features, and more too, as the seating faces can be reground many times, without removing the valve from its connecting pipe, before it is necessary to renew the seat or disc.

The "Renewo" is a valve of durability, not only because it is possible to renew all parts, but also owing to the scientific design of the seat and disc construction, which minimizes the wear on the seating faces, caused by the great velocity of the steam flowing through the valve,—insures perfectly clean seating faces, and therefore a tight valve,—and eliminates water-hammer.

Other features that aid in maintaining the high reputation of the valve, are,—a non-corrosive union connection between the hub and body, permitting the valve to be easily taken apart or assembled; a *renewable seat* made of a *hard* and *close-grained nickel,* having great durability,—and a stuffing-box that can be packed under pressure when the valve is wide open.

Lunkenheimer "Renewo" Globe, Angle and Cross Valves are made in sizes ranging from ¼ to 3 inches inclusive, and in two patterns,—Medium, for working pressures up to 200 pounds, and Extra Heavy, for pressures up to 300 pounds.

Specify and insist upon securing genuine *"Lunkenheimer make."* Do not accept substitutes, —they are never as good as the genuine.

"MOST supply houses sell them—yours CAN—if they DONT or WONT—tell *US*".

THE LUNKENHEIMER COMPANY

Largest Manufacturers of High Grade Engineering Specialties in the World

General Offices and Works, Cincinnati, Ohio, U. S. A.

NEW YORK,	CHICAGO,	BOSTON,	LONDON, S. E.,
64-68 Fulton Street	186 N. Dearborn Street	138 High Street	35 Great Dover Street

1

Do You Want To Reduce Boiler Maintenance Costs?

You know that scale incrustation increases fuel expense, results in excessive boiler wear and tear and high maintenance charges, and lessens the efficiency of the boiler.

Why not adopt a method which prevents scale formation? We have handled many thousand feed water problems for power plants; our chemists are feed water treatment experts, and our laboratories are equipped especially for this kind of work.

Each case is given individual consideration. That is why Dearborn Treatment accomplishes satisfactory results—absolute prevention of scale formation, corrosion, and other ill effects of the water, by the use of *treatment made to suit* the conditions. Write for catalog describing Dearborn methods.

Send gallon sample of your feed water for free analysis.

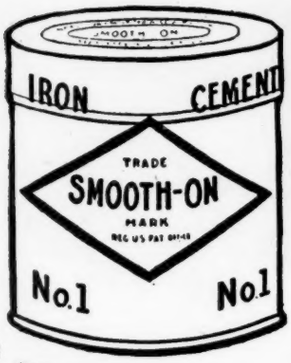
Send this today and we will give it immediate attention.

Dearborn Drug & Chemical Works

Robert F. Carr, President

General Offices, Laboratory and Works **General Eastern Offices, 299 Broadway**
Chicago **New York**

Branch Offices In Principal Cities

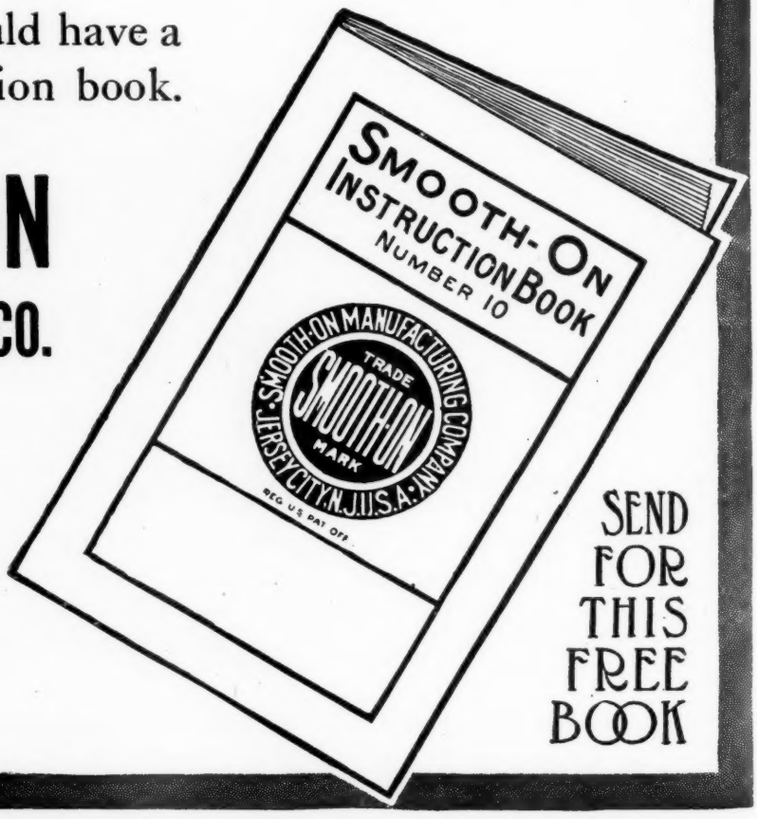


SMOOTH-ON

IRON CEMENTS

Positively stops all leaks of steam, water, fire or oil. They are easy to apply, harden quickly and when hard, expand and contract with the iron.

Every engineer should have a copy of our instruction book.



SEND FOR THIS FREE BOOK

SMOOTH-ON MANUFACTURING CO.

Jersey City, N. J.

231 N. Jefferson St., Chicago

36 Sacramento St.
San Francisco, Cal.

8 White St., Moorfields, E.C.
London

POWER

Vol. 35 NEW YORK, MARCH 19, 1912 No. 12

DID you ever go into a spick-and-span show power plant? Of course, you were impressed.

But didn't you come away with the main idea that much useless labor was spent in polishing brass work and cleaning white paint? Was not economy and efficiency pushed into the background?

A show plant is suitable for advertising purposes, but it is out of place when used for commercial work, unless enough help is employed to keep it in condition.

If an engineer has to polish useless brass work, he will not have the time necessary to devote to keeping his engines, boilers and auxiliaries in shape. No two ways about that. Might just as well expect him to obtain furnace economy with the firing doors open.

There are power-plant managers who bank more upon appearances than upon results. Beauty is but skin deep, and a boiler with a shining front and fittings may be so coated with scale that economy is out of the question. Don't forget that.

For instance, the "engineer" of a show plant went to a new job. He had kept the bright work polished and the machinery as clean as the proverbial whistle. The superintendent thought that this "engineer" was a "live one" and was sorry to see him go.

Another man came and in about four months he got fired.

Why?

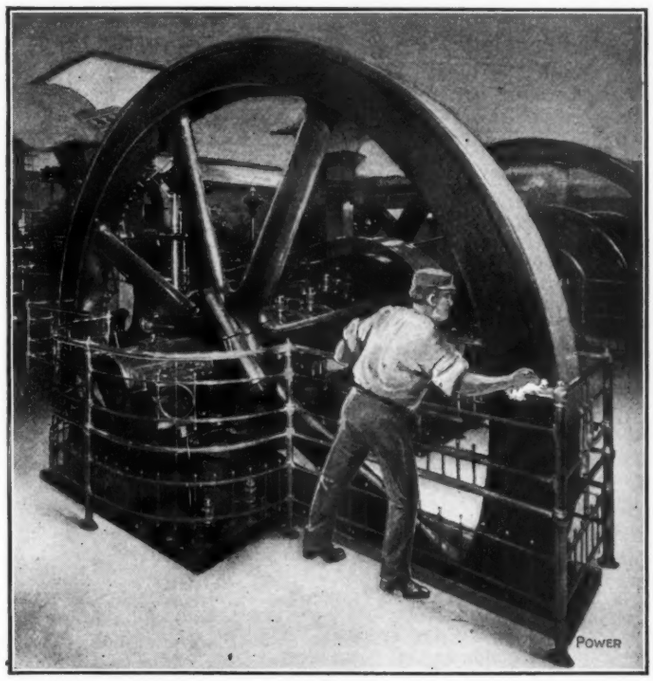
Because the first thing he did was to paint the brass work black and cut out useless work. He devoted his time to the engines and boilers. The superintendent thought he had used paint because he was too lazy to juggle brass polish and a cleaning rag.

But the engines were kept clean and ran smoother than they had with the former engineer. Instead of doing a wiper's work, this engineer put in his time stopping up air leaks in the boiler setting; he also removed over 7000 lb. of scale from the boilers.

A discarded feed-water heater that had not been used in six years was repaired and put in service again.

A comparison of indicator diagrams taken under old and new conditions showed that the new engineer had not been idle.

When this engineer found that he could not do all this useless cleaning and keep his engines and boilers in shape, he chose the lesser of two evils and cut out the brass work. He kept his machines in order and got discharged for doing what an engineer is paid to do.



A show plant was of more importance to the superintendent than well-kept machinery. Polishing rags and floor mops were given preference over the intelligent use of an indicator, or boilers free from scale.

A cheap man could have been hired to do the cleaning; for paying an engineer's wages to do wiper's work is not good business policy. When it comes to choosing between shining brass and economical operating conditions, cut out the glitter and give the engineer a chance to do an engineer's work.

An Interesting Power System

By C. A. Tupper

With sawmill refuse as fuel, steam is furnished to a turbine plant supplying power to the sawmill and is also conveyed through an exposed pipe line 2100 ft. long to a combination high-pressure engine and low-pressure turbine plant. The excess exhaust steam is used for heating.

An interesting combination of two modern steam-turbine plants, one supplied direct by boilers using sawmill refuse as fuel, and the other utilizing exhaust steam from an engine and auxiliaries supplied by the boilers of the first plant, is to be found at Virginia, Minn. Each of these stations, furthermore, is independent of the other, the steam in the latter instance being merely a commodity purchased. A general outline of the system is shown in Fig. 1.

The main generating plant is owned by the Virginia & Rainey Lake Co., a Duluth corporation engaged extensively in logging and transportation. At Virginia there are two large sawmills, located together, with the steam-turbine plant at one end. All wood waste from the mills, after being ground into small pieces, is conveyed with the sawdust (which is collected as shown in Fig. 2) to the boiler room and is fed continuously to the furnaces.

Boilers

There are fourteen 6x18-ft. return-tubular boilers, each of 150 hp., with dutch-oven settings and wood-burning furnaces. Steam from the boilers at 145 lb. pressure is collected in a 10-in. header and passes to the engine room through a 12-in. pipe, from which 5-in. branches lead to the turbines and also to a Murray Corliss mill engine.

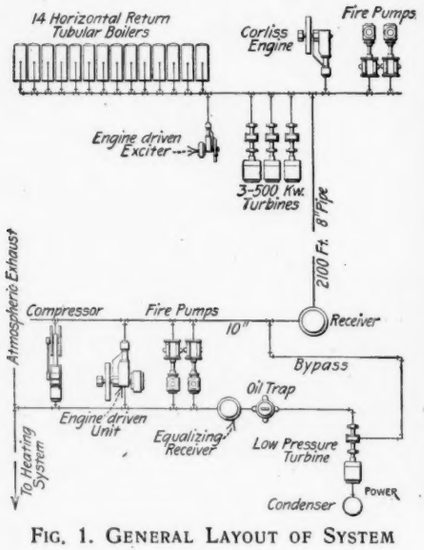


Fig. 1. General Layout of System

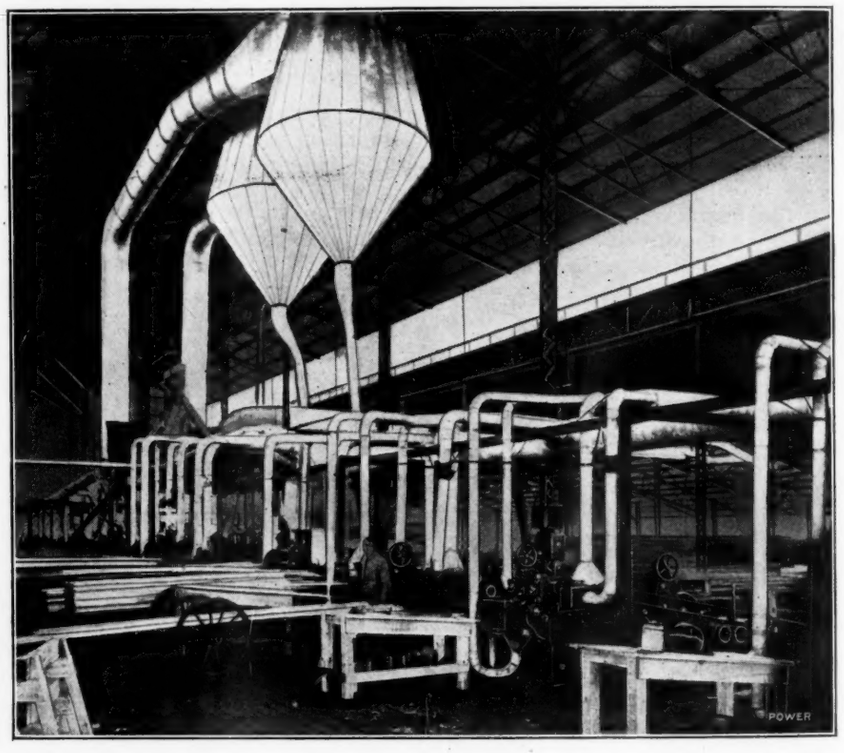


Fig. 2. Showing Method of Collecting Sawdust for Burning Under Boilers

Turbines

The turbines are three in number (see Fig. 3), each coupled to a 500-kw., three-phase, 60-cycle, 600-volt generator. Each turbine exhausts through a 24-in. nozzle to a Tomlinson condenser located directly underneath, and from this the water and air are removed by a rotary engine-driven pump. Injection water is taken from a cold well. The vacuum is maintained at about 28 in., and attached to the condenser head is an automatic vacuum breaker.

One 30-kw. engine-driven and one 35-kw. motor-driven generator furnish excitation current. There is also a 35-kw. motor-generator set for charging four electric yard locomotives.

Steam Distribution

Steam from the boilers of this plant is carried through an 8-in. pipe line 2100 ft. long (see Fig. 4), to the power station of the Virginia Electric Power & Water Co. This pipe rests on rollers supported by wooden posts, to each of which a 2-in. saddle has been strapped and at every 200 ft. there are expansion joints. The pipe is covered with 2-in. magnesia covering inclosed in ¼-in. painted canvas. The line ends in a suitable receiver and the drop in steam pressure over the entire distance does not exceed 3 to 5 per cent. Condensation is cared for by an automatic trap which discharges less than 10 gal. per hour, even in severe winter weather.

From the receiver the live steam passes to a 10-in. header and is distributed to an engine-driven generating set, of 300 kw. capacity, and to various other steam units, including an air compressor, and two fire pumps having a combined delivery of 3000 gal. per minute. The compressor operates the air lift from a well into a 275,000-gal. cistern, from which a steam pump raises the water to an overhead reservoir supplying the city mains.

There is held in reserve the original power unit of the company, consisting of two boilers and a George H. Corliss engine built over 40 years ago, which is belted to a 200-kw. generator.

Heating the Plant

The exhaust from all of the units is collected in a low-pressure line which, besides serving in winter to heat the power plant, the company's offices, two blocks distant, and other buildings in the vicinity, is used in a low-pressure Allis-Chalmers turbine. For the last-named purpose the steam passes first into a receiver, through an equalizing valve, and then, through an oil separator, at 16½ lb. absolute, into the turbine.

This machine is coupled direct to a 375-kw. alternating-current generator running at 3600 r.p.m., and operating in parallel with the engine-driven unit.

From the live-steam header there is a 2½-in. bypass by means of which high-pressure steam is automatically admitted when needed to the intermediate stage of the turbine. The pressure of this live steam varies from 145 to 80 lb., according to the demands of the sawmill power plant upon its boilers. The high-pressure connection is under control of the governor, so that, should the supply of low-pressure steam be inadequate, live steam will be admitted automatically to the turbine, this steam being throttled in its passage through the governor inlet valve.

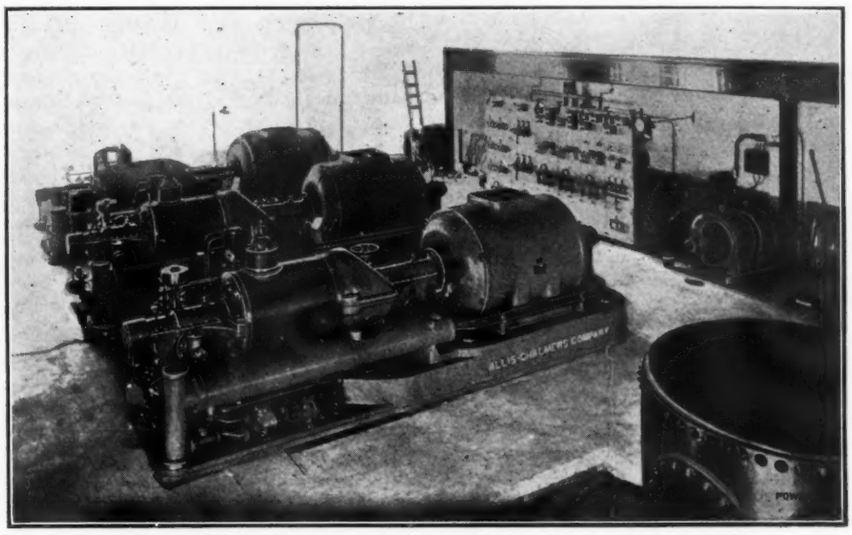


FIG. 3. THREE 500-KW. TURBO-GENERATORS

The air compressor and water pump are run continuously during the day and these units, with the condenser pump, supply enough exhaust steam for ordinary day loads. When the turbine load and the heating system do not call for the full quantity of steam exhausted, a relief valve in the exhaust receiver opens to the atmosphere; while, on the other hand, the heating system is insured of its full required supply in winter by a differential setting of the valves in the low-pressure system.

The turbine is kept going continuously and the engine-driven unit comes in on the late afternoon and evening load.

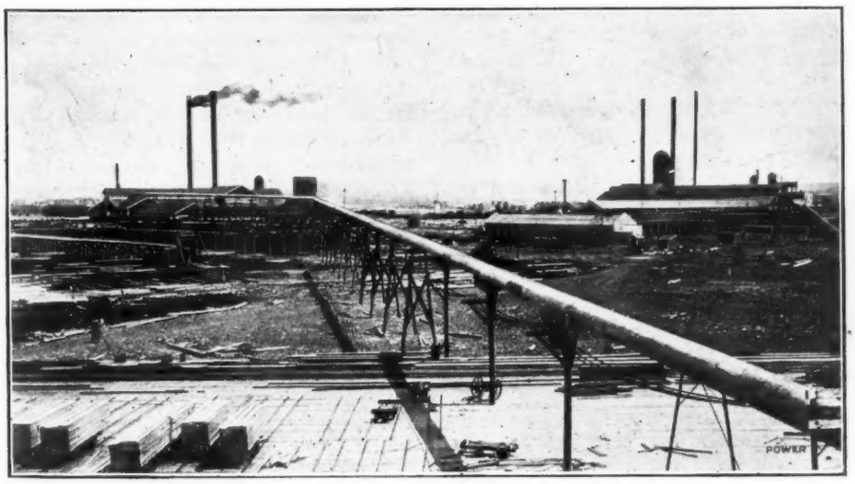


FIG. 4. PIPE LINE CARRYING STEAM 2100 FT.

Air Compressor Efficiency*

The development of compressors prior to the electrical era proceeded hand in hand with that of the steam engine, and its development has been marked by a gradual increase of rotary speed in order to keep pace, first with the steam engine and more recently with the electric motor. The latest development—that of the turbo-compressor—is one in which steam and electrical prime movers are equally interested, and very keen rivals.

Present-day methods of overcoming difficulties due to heating of the air during compression and of increasing the thermal efficiency of compressors are: (1) To water jacket the air cylinder and ports as thoroughly as their mechanical construction will allow. (2) To compress in stages if the final pressure required is so high as to make efficient lubrication difficult, the danger of spontaneous ignition of the lubricant (and frequently of entrained coal dust) imminent, or the power required for compression in one cylinder excessive.

In the days of slow-running machines, the valves were very simple, but as rotary speeds advanced the crude valves were found to be unsatisfactory, their weight alone being sufficient to cause the final pressure within the cylinder to rise considerably above that in the delivery main; hence it was recognized that for higher speeds improved valves were required. Present-day valves have resulted through development along the following lines: reduction of weight, reduction of lift, introduction of air buffers, use of auxiliary valve-gear, use of mechanically controlled valves, use of positively worked valves, introduction of unloaded valves and use of so called weightless valves.

Reduction of clearance volume, as time went on, was thought to be a matter of importance, and this gave Prof. Riedler an opening for the introduction of his mechanically controlled automatic valves. These were mounted upon the cylinder covers themselves, and thus reduced the clearance volumes to a minimum. The valve is, for moderate rotary speeds, an ideal one, and it is largely made use of even today on compressors of large size.

With the Riedler valve, compressor speeds advanced sufficiently to induce electrical engineers to advocate their direct driving by electric motors. For large units, the flywheel type of motor was usual, and its expense led to a further speeding up of the compressor. Thus, speeds were soon reached which made the elimination of extra valve-gear necessary; automatic valves again became popular, and extensive use was made of air buffers. But it was soon discovered that the air buffer not only acted to check the opening motion of the valve as desired, but it also hindered it in closing and thus led to its hammering on the valve seat. Buffers, therefore, gave way to the so called weightless type of valve, of which there are now innumerable varieties.

Turbo-compressors expose greater surfaces suitable for water jacketing than reciprocating compressors offer, and while being compressed in turbo-machines the air is distributed along passages and channels of smaller mean depth than those met with in the older type of

*Excerpts from paper read before the Rugby Engineering Society by B. M. Woodhouse.

machine. It appears; therefore, at first sight, that thorough cooling during compression should be an easy matter. Results scarcely confirm such expectation, chiefly because of two peculiarities of turbo-compressors, namely:

1. The amount of heat to be dissipated is considerably more than that due to adiabatic compression, as the mechanical energy lost through internal air leakage, unavoidable eddy currents, air impact, and air friction reappears as heat energy stored mostly in the air itself. Heat is also generated through the resistance opposed by the air to the motion of the rotors.

2. The temperature rise of the air is gradual; consequently the air temperature in the early stages of the compressor is too little different from that of the atmosphere to enable rapid dissipation through the necessarily somewhat thick cast-iron walls of the housing.

It can be said, Mr. Woodhouse continued, that since 1882 the rotary speed of compressors has gradually increased from 12 to 1200 r.p.m. in the case of reciprocating machines, and to 4000 r.p.m., and even more, in the case of turbo-compressors. This change has been accompanied in the case of reciprocating compressors by continual variation of valve design.

Heating of the air during compression is a source of waste of energy. A minimum of work is required to accomplish compression when the rise of pressure takes place isothermally, and, although with reciprocating dry compressors, this ideal is unattainable, it is still one after which the designer must aspire.

The author adds that if each stage of a compound reciprocating air compressor is considered as a distinct unit, it is safe to say that, in practice, compression takes place almost adiabatically. This is on account of the extreme difficulty of extracting from the air the heat generated during compression while the air is within the cylinder.

Briefly, the object of Mr. Woodhouse is to awaken interest (1) in air-valve design, as upon this detail the mechanical success of the air end of the unit is, to a large extent, dependent; (2) in the cooling of air during compression, because high thermal efficiency is certainly unattainable when water jackets only are used; (3) in the importance of volumetric efficiency, on account of the great effect it has upon the commercial efficiency of air-compressing machinery.

In his opinion, compressors can be run successfully and with good commercial efficiency at any speed suitable for reciprocating machinery of similar construction, provided always that the valves are of appropriate design, and that the air ports are constructed in such a way as to avoid excessive wire-drawing or too voluminous clearance spaces.

Chart of Factors of Evaporation

By Prof. Lionel S. Marks

In order to obtain a fair comparison of the performance of different steam boilers, it is necessary to measure their performance by some common scale. If we were to compare the lengths of several rods, we could do it either (1) by choosing one as a standard and expressing the lengths of all the others in terms of the length of that one; or (2) we could choose some arbitrary length (such as the foot) as unit of length and could then express the lengths of all the other rods in terms of that arbitrary unit. The latter procedure is followed in steam-boiler practice.

A steam boiler takes in heat from some external source, generally from a coal fire, and utilizes it for the evaporation of water. The efficiency of the boiler is most accurately measured by the percentage of the available external heat that it takes in. The usual method of measuring this amount of heat is by determining the weight of water evaporated per pound of coal burned under the boiler.

This method of measurement does not, however, permit a direct comparison between two boilers both of which are burning the same kind of coal. If, for example, one boiler takes in feed water at 210 deg. F. and generates steam at 90 lb. absolute and of a quality 0.98, and the other boiler takes in its feed water at 76 deg. and generates steam at 170 lb. absolute, superheat 150 deg., there will obviously be a considerable difference between the heat supplies per pound of steam generated in the two boilers. Actually, for the conditions just stated, the heat supplies would be in the ratio of 8 to 10, so that the evaporations per pound of coal in the two boilers would have to be in the ratio of 10 to 8 in order to get equal efficiencies from the two boilers.

In order to permit a true comparison of the efficiency of the boilers, it has become the general practice to reduce the actual weight of steam evaporated in the boiler to that which would have been evaporated, with the same amount of heat absorbed, if the feed water had gone to the boiler at 212 deg. and had been converted into dry and saturated steam at that same temperature.

According to the steam tables of Marks and Davis, the amount of heat required to evaporate 1 lb. of water under those conditions is 970.4 B.t.u. To find what weight of steam would have been evaporated in any given boiler with the actual heat that has gone into that boiler but with the assumed standard conditions of evaporation "from and at 212 deg. F.," it is necessary to find the actual heat supply per pound of steam formed in the boiler and to divide it by the heat supply (970.4 B.t.u.) per pound of steam in the standard boiler. This ratio is called the "factor of evaporation." If the actual weight of steam evaporated is multiplied by the factor of evaporation, we get the "equivalent evaporation" of the boiler; a quantity which permits a direct comparison between different boilers using the same kind of coal.

The accompanying chart permits the immediate finding of the factor of evaporation corresponding to any given condition of steam generation. The absolute steam pressure is plotted horizontally. The curves below the line marked "saturation line" are for wet steam, the quality (dryness factor) being marked on each curve. Above the saturation line, the curves are for superheated steam of the degree of superheat shown by the figures. The vertical scales on both sides of the chart give the corresponding factors of evaporation, on the assumption that the feed-water temperature is 32 deg. In practice, the feed-water temperature is always greater than 32 deg. F., and to correct for the actual temperature, something must be subtracted from the factor of evaporation as taken from the curves. The diagonal line permits of the immediate determination of this correction. The top horizontal scale is for feed-water temperature. The outside vertical scale on the left of the chart gives the corresponding correction for the factor of evaporation.

As an example of the use of this chart, we may take the conditions given earlier, namely, steam at 170 lb. absolute pressure, superheat 150 deg., with feed water at 76 deg. Starting with 170 lb. on the lower horizontal scale, and following vertically upward to the curve marked 150 deg., the intersection is seen to be on the horizontal line corresponding to the factor of evaporation 1.317. This, however, assumes a feed-water temperature of 32 deg.; the actual temperature is 76 deg. To correct for this, start from 76 deg. on the top horizontal scale and follow vertically down to the correction curve; the intersection is on the horizontal line corresponding to the correction 0.045 as given on the vertical scale on the extreme left. The actual factor of evaporation is consequently

$$1.317 - 0.045 = 1.272.$$

A patent was recently granted to E. A. Kelsey, Mt. Vernon, N. Y., for fitting valves to their seats by pressing one upon the other so that the imperfections in the shape of each will be reversely reproduced in the other, and guiding the valve to seat it in the same position in the seat in which it was fitted.

During 1911, the exports of British coal amounted to 64,599,266 long tons.

Notes on the Use of Belt Conveyers

By John J. Ridgway

Practical considerations covering the most efficient drives, feeds, arrangement of chutes, capacities and speed. The care of belt conveyers is also touched upon.

With level conveyers it makes very little difference whether the drive is at the head or tail, but when elevated a head-pulley drive should be used wherever possible, because driving from the tail brings the slack belt on the upper or load-carrying side to the detriment of the traction of the driving pulley and can only be compensated for by increasing the tension, which is undesirable.

With an intermediate drive where two snub pulleys are used in connection with the driving pulley, the greatest objection is the practical impossibility of getting the belt perfectly clean and especially in abrasive materials where the pieces are angular and often imbedded in the belt; this character of drive intensifies the injury resulting from the use of snub pulleys. These are always a detriment owing to the working side of the belt, when in its greatest tension, being forced against the face of the pulley.

It is too often the custom to minimize the head or driving pulley and increase the arc of contact by using the snub pulley. This is very bad practice, and in most cases can be eliminated by increasing the diameter of the driving pulley and making a long lead to the first return idler.

That type of troughing idler should be used which embodies the greatest simplicity in construction, is the most easily lubricated and does the least detriment to the belt.

The three-pulley setting—the center pulley horizontal and the two side pulleys at varying angles—is the simplest, cheapest, most easily lubricated and most efficient. In the writer's judgment, when the pulleys are in line the action of the horizontal pulley at the point of meeting with the side or angle pulley often produces the effect of a pair of shears; this can be eliminated, however, by offsetting the pulley.

Guide idlers should always be avoided, if possible. Their object is to control the shearing of the belt which, by reason of its not conforming to the outline of the idlers, and through imperfect contact or traction, has a tendency to run to that side where the traction is greatest, and if not controlled by these guide idlers would tend to run off.

This is so much the case that one often sees arguments used by belt manufacturers that they make special provision to protect the edge of the belt by more than the usual amount of cover or pad and draw attention to the perfection of attachment whereby this cover is vulcanized to the belt.

Traction versus Tension

The subject of traction ought to receive the greatest consideration, because a fabric when loose is not as easily cut as when in tension.

In many cases an undue amount of tension is placed upon conveyer belts; some due to lack of thought on the engineer's part, to penny-wise and pound-foolish practice, and to the necessity for employing tension in order to make the belt trough run approximately true.

The diameter of the head pulley is often reduced, thereby increasing the necessary tension; this practice saves a dollar in initial cost but may cost fifty dollars in wear and tear on the conveyer belt. Excess power due to tension or any other cause carries with it three detriments: First, the initial cost to furnish the power; the fixed charge for operation, and the wear and tear on belting and machinery occasioned by the extreme tension.

Feeds

Where material is fed to a belt these chutes should be so constructed and at such angles as to deliver the material at approximately the same speed at which the belt is traveling in order to produce the least possible abrasion. Moreover, the edge of the chute should be turned up slightly, to prevent the direct impact of the material on the belt. Conveyer belts are intended to convey, and not for impeding the velocity of the material; and this should always be kept in mind.

Sometimes it is desirable to feed a belt from a succession of bin gates with feeding chutes attached, in which case the principle underlying the initial feed chute should be embodied. It is very undesirable, although often done, to feed from two or more of these chutes at the same time; this is apt to overload the belt, and produce abrasion due to the discharge from a preceding chute being scraped and ground into the belt by its interference with a succeeding chute. Where multiple feed must be used mechanical control should be employed and not the usual type of gravity feed, which is always susceptible of the foregoing objection.

Chutes

Under this general heading are embodied initial feeding chutes, intermediate, transfer and tripper chutes, the latter often including a three-way chute. The great trouble in connection with all chutes is the disposition to minimize space to the detriment of good engineering practice and injury thereby to the belt. The latter being the most expensive item in connection with the conveyer, is the one which should receive foremost consideration, and the inevitable wear and tear thereof minimized.

In all chutes, whether direct or transfer, the material should be delivered to the center of the belt and the wings of the chutes should be lifting and set at an angle to the travel of the belt so that the material shall continually clear itself and not jam and grind under the edge of these wings.

Elevating

The subject of elevating material is one which is dependent on various contingencies and is controlled by a number of factors which must be individually considered, but the general principle is this: There is a relation between the traction of the material in connection with its angle of rest and velocity, that determines the angle of elevation.

These conditions are not always sufficiently considered, especially when they are complicated by various sizes of material, including small and large lumps, the percentage of each, the character of the feed, etc.

The question of intermittent or continuous feed is one which can be eliminated under almost all conditions because a continuous feed is more a question of theory than of practice.

Capacities

The disposition to figure the theoretical for the practical is too often the cause of disappointment, and it would seem hardly necessary to mention this if it were not overlooked so often in practice.

To find the capacity of a conveyer belt it is best to make a diagram of the troughing idler, measure 2 inches down from the edge of the belt, draw a straight line connecting these two points, figure the area therein, and calculate in tons, of whatever may be the particular character of material to be handled; discount this one-third and the result will as nearly approach the average maximum as one will get in practice.

Speed

The speed at which a conveyer belt can be run depends upon several factors, the character of the belt, the class of material and the spacing of the troughing

idlers. Originally, belts were run at speeds as high as 1000 feet a minute, but now the tendency seems to be toward that of slow speed, to the injury of clean deliveries, and consequent wear and tear to the belt at delivery and transfer chutes.

Deliveries

Where it is desirable that there should be the least possible breakage of material and where the distance from the conveyer to the bottom of the storage bin is 30, 40 or 50 feet, fish ladders can be used. These consist of boards or plates placed at angles to correspond with the material handled. Through this the material takes a zigzag course; or a center support is provided having a chute like a spiral staircase with a lip 1 or 2 inches high to prevent the material from being thrown off by centrifugal force. This carries the material to the bottom, spilling over the edge and building on itself until the bin is full.

Breakage

To eliminate breakage is not so much a question of speed as it is the character and design of the receiving and delivery chutes together with the general nature of the installation. Where conveyers are run in series each succeeding conveyer ought to be stepped up at least 5 per cent. in speed so that there shall never be any choking of the material in the transfer.

Trippers

Trippers, stationary or movable, should be used only where it is not possible to employ a mechanism that can be thrown into or out of operation, utilizing a counterweight to make this practical and looping the belt on the underside to give the necessary go and come and taking up the slack when the tripper is not in operation. All trippers embody the use of a snub pulley and they largely increase the tension, which is undesirable.

Automatic reversing trippers are complicated and expensive, and the writer has known only very few cases where the conditions made it possible to employ those embodying the automatic, reversing, continuously moving idea. In nine cases out of ten they are shifted from place to place, locked to the rail, and the automatic reversing feature is surplusage.

Cleaning a Belt

Sufficient care is not always taken to keep the belt and conveyer clean. This is frequently illustrated in plants where care is taken and the belts operate year after year giving satisfactory service, and a legitimate return in value or service rendered. The belt is relieved of the inevitable wear and tear contingent on the thoughtless, shiftless manner in which belt conveyers are frequently handled.

Decking

It is absolutely necessary to protect the lower line of belting from dust, grit and dirt that may fall upon it, for if any falls through on this lower line and goes on the pulley side of the belt it soon wears off the thin protective covering, bares the fabric and absorbs moisture.

Accidents often occur which are entirely avoidable, as, for instance, those coming from or originating in the initial feed, which in most cases can be provided for by a grating or screen of iron bars to catch any material liable to injure the belt.

Also damage occurs frequently through the careless use of the hoe, bar or shovel in cleaning out transfer chutes. Such instruments ought to be attached by chains or otherwise so that they cannot drop or catch in the belt, and before the power can be shut off, do almost irreparable injury.

Care should be taken in putting on a new belt to see that everything leads fair and true; for a set in a new belt is almost impossible to correct afterward.

Squaring the Ends

After a belt has been strung care should be taken before it is put in tension to see that it runs and lies true on the head, tail and upper and lower idlers, also through the stationary or movable trippers if such are in use.

A square should be placed on the belt, and the latter marked, using the long leg of the square along the edge of the belt.

Lacing or Fastening

In lacing a belt the edges should coincide and the middle of the belt should be fastened first; then put on the two outside fasteners (if a metal fastener be used), using one rivet on each side of each of the three fasteners. Tail the rivets from the joint so that the joints will draw together, and drive the rivets alternately, first on one side then on the other, opposite each other.

Starting and Stopping

Much harm is done by the thoughtlessness of those in charge of belt conveyers by not running their machines at full speed before they begin to feed material on them and also, especially at 12 and 6 o'clock, in the hurry to get away. They stop the machines too soon after the feed has ceased, and before the laggards or lumps which often roll back —advancing two steps and retreating one —have had time to be delivered.

This faulty method of starting and stopping is occasioned by the material dribbling over the head pulleys instead of being projected from them, catching in the transfer and other chutes. Although not a part of the engineering, this is something which can hardly be overlooked in their operation.

Conclusion

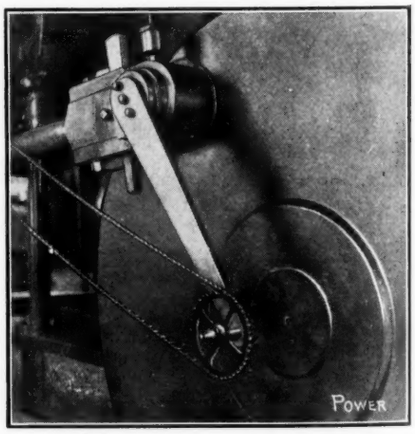
The abuse to which belt conveyers are so often submitted is on a par with the way the average human treats himself. The detriments that are insidious do not make themselves known in a smash-up or a breakdown, but they are nevertheless continuous, and eventually result in destroying the constitution of the man, the usefulness of the mechanism, and are the things which are so often disregarded.

In no type of machinery is it more necessary that constant care and common sense be employed than in the use of belt conveyers.

Peculiar Sprocket Wheel Arrangement

The sprocket wheel operating a telltale on a hoisting engine should be so placed as to be easily accessible in order to care for it and the sprocket chain.

There have been cases where the chain has come off the wheel and, running on the drum shaft, still rotate the telltale indicator, but behind its time. This condition in one instance resulted in wrecking a train of coal cars which were being hauled up a slope by running full speed into the foot of the shaft.



Sprocket Wheel Attached to Crank-pin by an Arm

On one engine the sprocket wheel was attached to a pin that was secured to the end of an arm which was bolted to the outer end of the crank pin, as shown in the accompanying illustration. The chain led to the indicator dial as shown. It is necessary to have the center of the sprocket wheel in line with the center of the crankshaft in order to insure true running.

What a Candidate for a Second Class License in Massachusetts Has to Know

By L. W. Chadwick

The following questions were asked an applicant for a second-class license in Massachusetts which limits the holder to having charge of 150 engine horsepower or less. However, he may operate a first-class plant under a chief engineer.

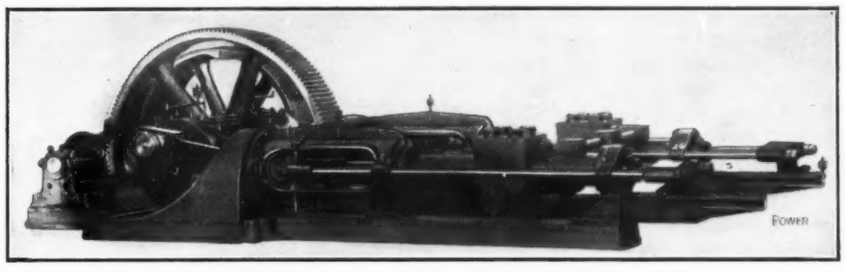
The applicant has made two attempts to get a second-class license within the last six months, and the following questions were repeated to the writer as being those asked by the examiners:

1. What is meant by horsepower in reference to a steam engine?
2. How do you figure the horsepower of an engine?
3. Give an example.
4. What is an indicator?
5. Describe an indicator.
6. What is it for?
7. Draw a good diagram from a single-eccentric Corliss engine.
8. How would you know whether it was from the high- or the low-pressure cylinder?
9. How late can a single-eccentric Corliss engine cut off?
10. Why not later?
11. Given the dimensions of a quadruple-riveted double butt-strap joint, find the efficiency of the joint.
12. Given the area to be braced, pressure carried, diameter and thickness of the shell, find the number of through braces 1 in. in diameter.
13. What parts of a locomotive boiler are braced and why?
14. Describe how each part is braced.
15. Given the dimensions of a water leg, the thickness of plate, pressure to be carried and size of stay-bolts, find how many are required.
16. Is grease detrimental to a boiler; if so, in what way or ways?
17. Where would you most expect to find a bag in a return-tubular boiler?
18. If a bag occurred in a girth seam, how would you patch it?
19. Is a bag dangerous; if so, why?
20. If a boiler has started to pit, what would you do to help stop it?
21. Describe the operation of an independent jet condenser. What is the office of the air pump?
22. Can such a condenser become dangerous to the engine? If so, in what way?
23. Why is scale in a boiler dangerous?
24. If you were forcing a battery of boilers, how could you tell by looking at the glass if one glass or column connection was plugged?
25. If, when you cut out a boiler by closing the stop valve, the pressure drops to zero (gage) and the water surges in the glass, what causes it to do so?
26. Given the number of square feet of heating surface in a boiler, how many square feet of grate surface are required for the same boiler?
27. If you found a bad lap crack, how would you repair it?
28. Where is the fusible plug in an Almy boiler?
29. In figuring the stress on a longitudinal seam, why is the radius used?
30. What is the use of an auxiliary piston in a steam pump?
31. How is it possible for a pump to work against a pressure greater than the steam pressure?
32. Draw a diagram and explain how to find the stress on 1 in. of a longitudinal seam. The same on a girth seam.
33. Given the dimensions, find the efficiency of a double butt-strap, double-riveted joint.
34. What is a double "D" valve and where is it used?
35. How is wood packing used and kept in place on a condenser with a bucket piston on the pump?
36. Given the number of square inches, thickness of plate, pressure to be carried, size of hollow stay-bolt, size of hole and stay-bolt, how many are required in the water leg of a locomotive type of boiler?
37. Find the pitch of rivets in a boiler. How do you find the size of them?
38. The safety valve on a boiler plays out and you get a new one; how would you know whether it was all right?
39. Would you recommend an open or a closed heater for factory work?
40. What is a horsepower constant?
41. If you take up on the spring of an automatic governor to speed up the engine and the latter races, what is wrong?
42. If an engine delivering 150 hp. suddenly drops to 140 hp., what is wrong and how may it be remedied?
43. If the steam pressure drops on a condensing engine, what can you do to help out until the steam pressure is up again?
44. How would you get a later cutoff on a single-eccentric Corliss engine?

5000 Pound Pressure Pump

Of more than ordinary interest is the large duplex, outside-packed, plunger type of power pumping engine shown in the illustration. This machine has four 2⅛-in. plungers, with a common stroke of 18 in., and at 47 r.p.m. of the crankshaft, is capable of delivering 50 gal. per minute against a working pressure of 5000 lb. per square inch. Two of these pumps were recently built for the Chase Rolling Mills, Waterbury, Conn., by the Deane Steam Pump Co., Holyoke, Mass.

The elimination of pressure joints in a machine of this type is a consideration of primary importance; consequently, in these machines, each cylinder is made integral with its valve chamber from a steel billet. The two chambers for the reception of the plungers and all internal water passages, subject to discharge pressure, are drilled from the solid metal. The valve service is of tool steel and each valve is separately accessible from above through its individual handhole opening; thus greatly facilitating the refacing of valves and seats.



General View of High-pressure Pump

The plungers being of small diameter, the stroke comparatively long, and the net plunger load high, special attention has been directed to the prevention of plunger deflection. The weight of the plungers, crossheads and side rods is supported at both ends of the cylinder on adjustable shoes, and in addition the alignment of the side rods is maintained by guides on the yoke and on the outer end of the cylinders.

The gearing is machine-cut from the solid blank, and the crankshaft, which operates in main bearings of the quarter-box type, is an openhearth steel forging. Annealed-steel castings are used for the disks which are integral with the pins, and the connecting-rods are of the solid-end type, of machine steel, fitted with phosphor-bronze bearing boxes and made adjustable for wear at both ends. The crossheads are fitted with adjustable shoes and are operated in bored guides in the yokes. Drippings of lubricating oil and stuffing-box leakage on the foundation are prevented by deep retaining flanges and pans, made integral with the machine.

Schedule of Flanges, Flanged Fittings

In the June 27, 1911, issue was printed a schedule of flanged fittings arranged by the committee on standardization of the American Society of Mechanical Engineers, acting in conjunction with a similar committee from the National Association of Master Steam and Hot Water Fitters.

Since this earlier report of the committees, however, the schedule has been revised and considerably amplified, and in its present form, as herein given, has been recommended officially by the American Society of Mechanical Engineers, and adopted by the National Association of Master Steam and Hot-Water Fitters. Furthermore, at the annual meeting of the American Society of Heating and Ventilating Engineers the new schedule was recommended for use by its members.

The personnel of the committees on standardization was as follows:

American Society of Mechanical Engineers: H. G. Stott, John H. Sparrow, William Schwanhausser, Walter M. McFarland, Albert C. Ashton.

National Association of Master Steam and Hot Water Fitters: William J. Baldwin, Jr., N. Loring Danforth, William T. English, William P. Kirk, Thomas B. Cryer, Edward B. Denny, William H. Oakes.

Explanatory Notes

The following explanatory notes will be found useful in connection with the schedule:

1. Standard or extra-heavy reducing elbows carry the same dimensions center to face as the regular elbows of the largest straight size.

2. Standard or extra-heavy tees, crosses and laterals, reducing on run, carry the same dimensions face to face as the largest straight size.

3. If flanged fittings for lower working pressures than 125 lb. are made, they shall conform in all dimensions, except in thickness of shell, to this standard, and shall have the guaranteed working pressure cast on each fitting. Flanges for these fittings must be of standard dimensions.

4. Where long-turn fittings are specified, it has reference only to elbows, which are made in two center to face dimensions, to be known as "elbows" and "long-turn" elbows, the latter being used only when so specified.

5. All standard-weight fittings must be guaranteed for 125 lb. and extra-heavy fittings for 250 lb. working pressure, and each fitting must have some mark cast on it indicating the maker and the guaranteed working steam pressure.

6. All extra-heavy fittings and flanges to have a raised surface $\frac{1}{16}$ in. high inside of the bolt holes for the gasket.

Standard-weight fittings and flanges to be plain faced.

Bolts to be $\frac{1}{8}$ in. smaller in diameter than the bolt holes.

Bolt holes should straddle the center lines.

7. Size of all fittings scheduled indicates the inside diameter of the ports. For the outside diameter of pipe use the corresponding size of the inside diameter fittings.

8. The face to face dimension of a reducer, either straight or eccentric, shall be equal to the diameter of the larger flange.

9. Square head bolts with hexagonal nuts are recommended.

10. Twin ells, double branch ells, side outlet ells, side outlet tees and four-way tees, whether straight sizes or reducing, carry the same dimensions center to face and face to face as the regular ells and tees.

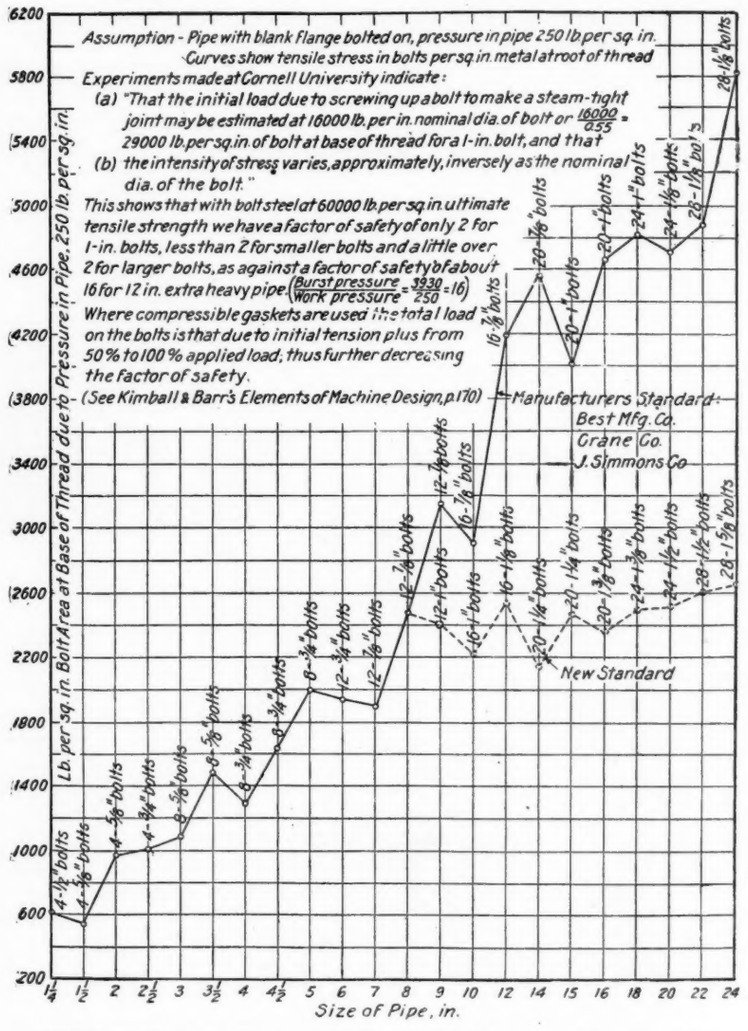
11. Bull-head tees or tees increasing on outlet will have the same center to face and face to face dimensions as a straight fitting of the size of the outlet.

12. Up to and including the 4-in. size, center to face and face to face dimensions of reducing fittings will be the same as that of a straight fitting of the larger opening.

13. Pipe sizes 14 in. and over refer to outside diameter.

Stresses in Bolts

In connection with the report of the standardization committee, the February issue of the *Journal* of the American Society of Mechanical Engineers contains the accompanying chart, showing the stresses in the bolts at the base of the thread due to a pressure in the pipe of 250 lb. per square inch. The chart shows the extent to which these stresses increase with the sizes of the pipe in the Manufacturers' Standard, and the dotted lines show the departure made from the latter in order to keep the stresses down to what is considered a safe limit.


Assumption - Pipe with blank Flange bolted on, pressure in pipe 250 lb. per sq. in.
Curves show tensile stress in bolts per sq. in. metal at root of thread
Experiments made at Cornell University indicate:
(a) "That the initial load due to screwing up a bolt to make a steam-tight joint may be estimated at 16000 lb. per in. nominal dia. of bolt or 16000/0.55 = 29000 lb. per sq. in. of bolt at base of thread for a 1-in. bolt, and that
(b) the intensity of stress varies, approximately, inversely as the nominal dia. of the bolt."
This shows that with bolt steel at 60000 lb. per sq. in. ultimate tensile strength we have a factor of safety of only 2 for 1-in. bolts, less than 2 for smaller bolts and a little over 2 for larger bolts, as against a factor of safety of about 16 for 12 in. extra heavy pipe (Burst pressure / Work pressure = 3930/250 = 16)
Where compressible gaskets are used the total load on the bolts is that due to initial tension plus from 50% to 100% applied load; thus further decreasing the factor of safety.
(See Kimball & Barr's Elements of Machine Design, p. 170)
Manufacturers Standard
Best Mfg. Co.
Crane Co.
J. Simmons Co.
New Standard
Lb. per sq. in. Bolt Area at Base of Thread due to Pressure in Pipe. 250 lb. per sq. in.
Size of Pipe, in.

Stress at Base of Thread in Bolts for Different Sizes of Pipe

SCHEDULE OF FLANGES AND FLANGED FITTINGS

Standard Weight Flanged Fittings

Size, Inches	All Fittings and Flanges						90° Elbow	45° Elbow	Long Turn Elbow	Tee	Cross	Lateral	Y Branch			"Y"	
	Diameter of Flange	Thickness of Flange	Number of Bolts	Diameter of Bolt Circle	Diameter of Bolt Holes	Size of Bolts	Center to Face "C"	Center to Face "C"	Center to Face "C"	Center to Face "C"	Face to Face "F"	Center to Face "C"	Center to Face on Short End "C_1"	Center to Face of Outlets	Face to Face "F"	Center to Face "C"	Center to Face Short End "C_1"
1	4	7/16	4	3	5/16	7/16	3½	2	5½	3½	7	6	2	...	8	3½	2
1¼	4½	½	4	3⅜	⅜	7/16	3¾	2	6	3¾	7½	6¼	2¼	...	8½	3¾	2¼
1½	5	9/16	4	3¾	⅜	½	4	2¼	6½	4	8	6¾	2⅜	...	9¼	4	2⅜
2	6	⅝	4	4¾	¾	⅝	4½	2½	7	4½	9	8	2½	...	10½	4½	2½
2½	7	11/16	4	5½	¾	⅝	5	2¾	7½	5	10	9½	2½	...	12	5	2½
3	7½	¾	4	6	¾	⅝	5½	3	8	5½	11	10	3	...	13	5½	3
3½	8½	13/16	4	7	¾	⅝	6	3½	9	6	12	11½	3	...	14½	6	3
4	9	15/16	8	7½	⅞	¾	6½	4	10	6½	13	12	3	...	15	6½	3
4½	9½	15/16	8	7¾	⅞	¾	7	4½	11	7	14	12½	3	11⅞	15½	7	3
5	10	15/16	8	8½	⅞	¾	7½	4½	12	7½	15	13½	3½	12½	17	7½	3½
6	11	1	8	9½	⅞	¾	8	5	13	8	16	14½	3½	13	18	8	3½
7	12½	1 1/16	8	10¾	⅞	¾	8½	5½	14½	8½	17	16½	4	13⅞	20½	8½	4
8	13½	1⅛	8	11¾	⅞	¾	9	6	16	9	18	17½	4½	14⅞	22	9	4½
9	15	1¼	12	13¼	⅞	¾	10	6½	18	10	20	19½	4½	17¼	24	10	4½
10	16	1 3/16	12	14¼	1	⅞	11	7	20	11	22	20½	5	18⅛	25½	11	5
12	19	1¼	12	17	1	⅞	12	7½	22	12	24	24½	5½	20	30	12	5½
14	21	1⅜	14	18¾	1⅛	1	14	7½	24	14	28	27	6	21½	33	11	6
15	22¼	1⅜	16	20	1⅛	1	14½	8	26	14½	29	28½	6	22⅛	34½	14½	6
16	23½	1 7/16	16	21¼	1¼	1	15	8	28	15	30	30	6½	22⅞	36½	15	6½
18	25	1 9/16	16	22¾	1¼	1¼	16½	8½	30	16½	33	32	7	24½	39	16½	7
20	27½	1 11/16	20	25	1¼	1¼	18	9½	32	18	36	35	8	25⅞	43	18	8
22	29½	1 13/16	20	27¼	1¼	1⅛	20	10	34	20	40	37½	8½	27⅛	46	20	8½
24	32	1⅞	20	29½	1¼	1⅛	22	11	36	22	44	40½	9	28½	49½	22	9
26	34½	2	24	31¾	1⅜	1¼	23	13	39	23	46	43½	9½	30⅞	53	23	9½
28	36½	2 1/16	28	34	1⅜	1¼	24	14	42	24	48	46½	10½	33	57	24	10½
30	38¾	2⅛	28	36	1½	1⅜	25	15	45	25	50	49½	11	35½	60½	25	11

Extra Heavy Flanged Fittings

1	4½	11/16	4	3¼	⅜	½	4½	2½	5½	4½	8½	7½	2⅜	...	9½	4½	2⅜
1¼	5	¾	4	3¾	⅜	½	4½	2½	6	4½	8½	7½	2⅜	...	9½	4½	2⅜
1½	6	13/16	4	4½	¾	⅝	4½	2¾	6½	4½	9	8½	2½	...	11	4½	2½
2	6½	⅞	4	5	¾	⅝	5	3	7	5	10	9½	2½	...	12	5	2½
2½	7½	1	4	5⅞	⅞	⅝	5½	3¾	7½	5½	11	10½	2½	...	13	5½	2½
3	8½	1⅛	8	6⅝	⅞	¾	6	3½	8	6	12	11½	3	...	14½	6	3
3½	9	1 3/16	8	7¼	⅞	¾	6½	4	9	6½	13	12½	3	...	15½	6½	3
4	10	1¼	8	7⅞	⅞	¾	7	4½	10	7	14	13½	3	...	16½	7	3
4½	10½	1 5/16	8	8½	⅞	¾	7½	4½	11	7½	15	14½	3½	13⅜	18	7½	3½
5	11	1⅜	8	9¼	⅞	¾	8	5	12	8	16	16	3½	14	19½	8	3½
6	12½	1 7/16	12	10⅝	⅞	¾	9	5½	13	9	18	17½	4	14⅝	21½	9	4
7	14	1½	12	11¾	1	⅞	9½	5¾	14½	9½	19	19	4½	15⅜	23½	9½	4½
8	15	1⅝	12	13	1	⅞	10½	6	16	10½	21	20½	5	16	25½	10½	5
9	16¾	1¾	12	14	1¼	1	11	6½	18	11	22	22¾	5	20⅞	27¾	11	5
10	18½	1⅞	16	15⅜	1¼	1	12	7	20	12	24	25½	5½	21⅝	30¾	12	5½
12	20¾	2	16	17¾	1⅜	1⅛	13½	8	22	13½	27	27¾	6	23	33¾	13½	6
14	23½	2⅛	20	20½	1⅜	1⅛	15	9½	24	15	30	31½	6½	24⅜	37¾	15	6½
15	25	2 3/16	20	21½	1⅜	1⅛	15½	10	26	15½	31	33½	6½	25⅛	39¾	15½	6½
16	26	2¼	20	22½	1½	1⅛	16½	10½	28	16½	33	34½	7½	25⅞	42	16½	7½
18	28½	2⅜	24	24¾	1½	1⅛	17¾	11	30	17¾	35½	37¾	8	27⅛	45¾	17¾	8
20	31	2½	24	27	1⅝	1½	19½	11½	32	19½	38½	40¾	8½	28⅝	49½	19½	8½
22	33	2⅝	28	29½	1⅝	1½	20½	12	34	20½	41	43¾	9½	30	53½	20½	9½
24	36	2¾	28	32	1¾	1⅝	22½	13	36	22½	45	47½	10	31½	57½	22½	10

Condensers for the Power Plant

By William F. Fischer

General classification of condensers, including the jet, barometric, siphon ejector and surface types; description of various types; their operations, advantages and limitations.

Condensers may be divided into two general classes: jet and surface condensers. To the former class belong barometric, siphon, ejector, spirojector, induction and, in fact, all types in which the cooling water mingles with the steam. Under surface condensers are included all types in which the cooling water is separated from the steam by metallic surfaces either in the form of tubes or plates.

Jet Condensers

In this type the steam is usually admitted at the top of the condenser chamber and the water outlet is at the bottom so that the condensed steam and cooling water will drain into the wet-vacuum pump. On account of the vacuum in the condenser preventing the pressure of the atmosphere from acting upon the surface of the water, it is impossible for any suction pump to lift the water out of the condenser; hence the water must flow to the suction side of the pump by gravity. If the cooling water is to be used for boiler feed after leaving the condenser, it should discharge to a hotwell, the grease and oil being extracted before sending it to the boilers, or the exhaust steam may be passed through an oil separator before entering the condenser.

If an open feed-water heater and purifier are used the water is drawn from the hotwell by low-service pumps and discharged to the heater, from which the heated water flows to the inlet of the feed pumps by gravity. If a closed feed-water heater is used, however, the feed pumps draw their supply direct from the hotwell and force it through the closed heaters which are located on the high-pressure side of the pumps.

If the cooling water is unsuitable for boiler feed, on account of scale-forming impurities, it should be thoroughly treated before using, or it may be wasted altogether. In the latter case, however, the distilled water represented by the condensed steam is also wasted and as a consequence this may represent a considerable loss in dollars and cents each year. It might be better in such a case to use a surface condenser. If the cooling water is suitable for boiler feed, however, a jet condenser is unexcelled and should be given preference over the surface condenser because of its simplicity, low first cost, low cost of attendance, repairs, etc. Furthermore, they occupy less floor space, the erection charges are considerably lower and less cooling water is required.

Jet condensers should always be provided with a device for automatically breaking the vacuum by admitting air when the hight of water in the condenser reaches a certain point. Another method of preventing the condensing water from backing up in the engine cylinder is to provide a check valve in the exhaust pipe between the engine and the condenser.

Every condensing plant should be equipped with a free exhaust valve, so placed as to permit the exhaust steam from the engines to pass freely to the atmosphere in case the condenser fails to operate, or is shut down for repairs. The design of these valves varies somewhat with different manufacturers, but in general they are of the mushroom type, the valve being partly balanced by a lever and weights, or by weight and springs combined. These valves should be water-sealed to prevent air leakage between the seat and the disk.

Countercurrent Jet Condensers

Jet condensers of the countercurrent type are found to give much better results than the parallel-flow type and are more generally used. The injection water entering the condenser at one end and the exhaust steam at the other insure the coolest water coming in contact with the coolest steam, and the hottest steam coming in contact with the hottest water. Thus it is possible to raise the temperature of the condensing water very close to the temperature of the exhaust steam before discharging it from the condenser. This results in a considerable saving in the quantity of cooling water and also insures a much higher temperature for the boiler feed if drawn from the condenser hotwell.

In jet condensers of the parallel-flow type in which the water and steam enter at the same end and flow in the same direction, the hottest steam comes in contact with the coldest water, and the coolest steam comes in contact with the hottest water; hence the temperature of the discharge water may be some 10 to 20 degrees cooler than that of the exhaust steam entering the condenser, or the temperature corresponding to the vacuum.

Even though plenty of cold water is available for condensing purposes, it will be found advisable to work the condensing water as close to the temperature limit as possible in order to reduce the pumping charges to a minimum and keep the hotwell temperature as high as possible, at the same time reducing the work of the dry-air pump, as less air is brought in with the condensing water.

The amount of air to be handled in the case of a jet condenser is greater than with the surface condenser, due to the air carried in with the cooling water.

Barometric Condensers

Jet condensers in which the wet-vacuum pump is eliminated entirely, and the cooling water and condensed steam are discharged by gravity to the hotwell, are known as barometric condensers.

The features of this type are the same as mentioned under the general classification of jet condensers with the exception that the lower end of the condensing chamber is placed 34 feet or more above the surface of the water in the hotwell, the cooling water and condensed steam discharging into a vertical tail pipe, the lower end of which is submerged some 3 feet or more below the surface of the hotwell water, the latter forming a seal which effectively prevents air from leaking back into the condenser through the end of the pipe. Care should be taken to see that there is more water above the bottom of the tail pipe than will fill the tail pipe up to the bottom of the condenser casting; otherwise, when starting up, the water may be drawn back into the tail pipe, uncovering its end and admitting pockets of air which would tend to break the vacuum.

In condensers of the barometric type there is no danger of the condensing water backing up into the engine cylinders when the circulating pump stops, as the column of water in the tail pipe is too heavy to be forced back the full hight of 34 feet by the pressure of the atmosphere. When the circulating pump stops and the exhaust pressure builds up in the condenser, the atmospheric relief valve opens and the engine runs non-condensing until the circulating pump starts again. There is only one danger in flooding a condenser of this type when properly designed and erected, and that is if more water is pumped to the condenser than will flow freely through the tail pipe.

Another advantage of the barometric type is that it has no connection to the suction side of a wet-vacuum pump and therefore a sudden overload on the engine does not reduce the supply of condensing water at a period when there is the greatest demand for it.

The general design of barometric condensers varies considerably with different makes, and they are built on both the parallel-flow and the countercurrent principles.

Siphon or Injector Condensers

In barometric condensers of the siphon or injector type the dry-air pump may be eliminated entirely, the only pump necessary being a circulating pump, to lift the water to within about 18 or 20 feet of the condenser-inlet pipe, the vacuum taking care of the upper 18 or 20 feet of the lift. In condensers of this type the condensed steam, air and noncondensable vapors are carried out through the tail pipe with the condensing water.

If the level of the injection or cooling-water supply is not more than 18 or 20 feet below the condenser inlet, these condensers will siphon the water over as soon as a vacuum is formed, and in this case the circulating pump will not be required, the condenser operating without pumps of any kind whatsoever, making the cheapest and simplest possible arrangement. If the condenser is to siphon its water supply in this manner a cross connection should be made between the injection pipe and the tail pipe at the supply level. This connection should be provided with a gate valve for admitting water to the tail pipe when starting up; the water falling through the tail pipe into the hotwell draws the air out of the condenser cone and exhaust pipe, thus forming enough vacuum to draw the injection water up to the condenser inlet. The starting valve is then closed and the water siphons over, due to the difference in water levels and to the vacuum in the condenser. When starting the condenser it is a good plan to run the engine very slowly in order to have a small quantity of exhaust steam flowing to the condenser to assist in creating a vacuum to siphon over the water in the injection pipe. Where there is no head water available the circulating pump must be used but will have to work against only a low head.

The many different types of siphon or injector condensers, spirojector condensers, etc., all work on the same principle. The cooling water enters the condenser on the side through an annular ring surrounding a hollow cast-iron exhaust cone with the steam inlet at the top. The injection water passes downward in a thin film around the outside of the exhaust cone, and the exhaust steam flowing down inside the cone comes in contact with the falling water and is condensed.

The volume of cooling water and condensed steam falling vertically through a contracted neck in the condenser tail pipe is greatly increased in velocity at this point and the air and noncondensable vapor are drawn out of the condenser with the rapidly falling column of water, the contracted neck taking the place of an air pump.

In some types, the hollow cast-iron cone may be raised or lowered to regulate the supply of condensing water. The only danger of flooding such a condenser would be if the cone was so adjusted as to admit more water than would pass freely through the contracted neck and the tail pipe of the condenser, in which case the water would back up into the exhaust pipe.

Condensers of this type are of relatively low cost as compared with other types, and very economical when water can be obtained from a reservoir or stream at such an elevation as to make pumping unnecessary.

Ejector and Induction Condensers

In condensers of the ejector and induction types the condensing water enters at the top and passes downward through special forms of cone-shaped combining nozzles in a solid jet and at very high velocity, the momentum of the flowing water ejecting the discharge without the aid of a pump. No air pump is required with this condenser as the air and noncondensable vapors are carried away with the ejection water by the inductive power of the jet. The exhaust steam enters through the side of the condenser and passes downward through a series of inclined orifices or nozzles at high velocity. Upon coming in contact with the jet of cooling water flowing in the inner chamber, it is condensed, thus imparting the velocity of the steam direct to the cooling water.

The condenser is so arranged that it may be started by a supplementary jet of steam or water, after which it will siphon over its own water, providing the lift is not over 16 feet. Once the jet is started the cooling water is drawn in continuously by virtue of the vacuum formed and sufficient velocity is thus imparted to the jet to discharge the combined mass of condensed steam, cooling water and air against the pressure of the atmosphere. This condenser, like the others, should be provided with an automatic free-exhaust valve. To prevent water from backing up into the engine cylinder these condensers are provided with a check valve in the exhaust pipe.

Such a condenser is limited in operation as it lifts its own injection water by the velocity imparted to the jet by the exhaust steam and the vacuum formed. If the condenser is not correctly proportioned for the engine or if the load on the engine is variable, there may be times when the small amount of exhaust steam to be condensed will not be sufficient to impart the necessary velocity to the injection water and the condenser will not operate satisfactorily. Where the condenser takes its water under pressure, however, this objection does not apply, for in this case the head or pressure of the water determines the velocity of the jet independently of the volume of exhaust steam entering the condenser.

Surface Condensers

The surface condenser as usually constructed consists of a cast-iron shell into which a large number of metallic tubes of from ⅝ inch to 1 inch in diameter are placed and secured at each end into a tube sheet. These are so arranged with connecting chambers that the cooling water flows through the inside of the tubes while the exhaust steam impinges upon the outside and is condensed upon striking the cooled surfaces.

In these condensers provision should always be made to insure free expansion and contraction of the tubes into the tube sheets. This is usually accomplished by passing one end of each tube through an adjustable packing gland which permits a free movement of the tube during expansion and contraction and at the same time prevents leakage between the tube and header; the other end of the tube is secured rigidly to the tube sheet. The tubes are usually made of solid-drawn brass or copper, or an alloy composed of copper, zinc and tin, and vary in thickness from No. 16 to No. 18 B. W. gage. The tube plates or headers into which the tubes are secured are usually made of the best rolled brass or muntz metal.

The outer shell or casing is made of cast iron in one or more sections bolted together. Baffle plates are usually attached to the inside of the condenser directly opposite the steam inlet to assist in distributing the steam more evenly over the tube surface. To prevent excessive flooding of the lower tubes in a surface condenser the upper tubes should be so arranged and spaced as to permit the steam to penetrate freely to the lower rows, thus insuring a more uniform cooling effect upon all the tubes.

One of the advantages of the surface condenser is that the exhaust steam, being separate from the condensing water, the condensate may be used over and over again in the boilers even though dirty or corrosive water be used for condensing purposes. There is no danger of the condensing water entering the engine cylinder, and even if the condenser becomes entirely flooded by the condensed steam, due to a failure of the wet-vacuum pump, there would be no danger of cylinder flooding as the accumulated pressure of the steam would force open the atmospheric relief valve and would allow the engines to exhaust to the atmosphere.

One objection to the surface condenser is that the grease from the engine cylinders, which is carried over by the exhaust steam, accumulates on the inside of the tubes and reduces the efficiency of the cooling surface. If the exhaust steam is first passed through an efficient grease extractor or separator this may be prevented to a large extent.

For a given capacity, surface condensers are high in first cost and are the most expensive to keep in good working condition, due to the frequent necessity for cleaning, renewing defective tubes, etc. Where boiler-feed water is a very important item, however, surface con-

densers represent a considerable saving, as nearly all the condensed steam is returned to the boilers as distilled water, free from scale-forming substances, and the saving thus effected will in most cases more than offset the additional cost of this type of condenser.

Low Pressure Fans

By John S. Leese

The use of low-pressure air fans has been rapidly increasing during the past few years. The field for their application is wide, covering as it does nearly every class of engineering work. The causes for this recent general adoption of the low-pressure air mover are not far to seek; they may be traced to recent legislative regulations regarding the ventilation and atmospheric conditions of industrial establishments; to the increased need for "hustling" in manufacturing necessitating resort to the artificial drying of goods and other time-saving propositions; to the realization by manufacturers that the absence of contaminating factors in the atmosphere of their factories and workshops militates toward the production of higher-quality goods; and to the fact that many latter-day tendencies—as, for instance, forced boiler draft—call for simple and efficient methods for the introduction of low-pressure air.

Two Types

The fans herein considered may be either of two types, the propeller type or the regular centrifugal type. The former class handles large volumes and low pressures, while the latter deals with higher resistances and speeds of discharge. The volume and discharge velocity of both types vary directly as the revolutions per minute; the pressure as the square, and the power required to drive the fan as the cube of the revolutions. In the case of the propeller type, the revolutions being constant, the volume of air discharge varies as the square root of, and the pressure inversely as, the discharge area; while the power required is least with a wide-open discharge area, increasing to about three times this amount as the area is closed. As to the centrifugal type, the volume varies directly with the area of discharge, the pressure remains practically constant throughout the whole range of area, and the power required to drive the fan increases with the volume of air handled.

Design

The following rules are important in the design of fans: (1) The blades must not be designed in such a way that eddy currents will be engendered; (2) the leading edge of the blades must offer no obstruction to the influx of the air handled; (3) the discharge edge of the blades must be designed with a similar object in view, but care must also be taken that no unnecessary velocity is imparted to the air at the discharge opening; (4) the blades must be made as narrow as possible in the direction of air travel; (5) no obstructions must be placed in the path of the air in its travel through the fan; (6) owing to the comparatively low linear velocity of the hub ends of the blades, these must be so designed that the accumulation of air pressure in the vertical plane behind the fan cannot force any air through the fan in the wrong direction in the vicinity of the hub. Adherence to these conditions will tend toward the production of a satisfactory high-efficiency fan.

Three Kinds of Efficiency

There are three kinds of efficiency to be considered in connection with fans —mechanical, volumetric and manometric. The mechanical efficiency indicates the ratio of the potential power in the air after it has been passed through the fan to the power expended in driving the fan. The volumetric efficiency refers to the ratio between the volume of air actually handled and to the theoretical volume of air which should be handled, if slip were eliminated. Manometric efficiency is the relation between the pressure actually obtained with a fan and the pressure equivalent to the highest speed of the fan. The importance of a good mechanical efficiency is obvious while volumetric efficiency is an important consideration in low-pressure, high-volume fans; that is, the propeller type of fans. Manometric efficiency is not of great importance with the propeller type, but with the centrifugal type, as the peripheral speed increases, so does the necessary strength and hence the cost of manufacture; therefore manometric efficiency is important in designing this type of fan.

Conditions of Application

The choice of a fan depends upon the application to which it is to be put. The air piping system and distributing ducts are for this reason designed, and very often installed, first. The following conditions may be applied to all air systems, for whatever purpose, since their fulfilment must be beneficial to a greater or less degree in each case. The air ducts should be as short as possible, large in cross-sectional area and internally smooth, with as few changes of direction as practicable; such changes in direction as are necessary should be effected with easy bends, with any necessary variations in the cross-sectional area made gradually and as air-tight as possible, especially so in the neighborhood of the fan.

Type of Construction

Having laid out the system, the next step is the purchase of a suitable fan. The type of construction depends upon the relation between the volume to be delivered and the pressure to be maintained. For instance, low-pressure fans of large capacity should have large air-ways, the speed being allowably low in comparison to that of fans designed for higher pressures and smaller volumes, the latter having comparatively small air-ways.

As a general rule, it may be said that for free air circulation the propeller type of fan is most suitable for pressures up to about ¼-in. of water, while from ⅜-in. of water upward the use of centrifugal fans may be advantageous; between these pressures the choice is not to be so definitely stated. The considerations of quietness in running and low power are in favor of the centrifugal type fan; while, from the view point of first cost and the allowable speed limit due to the construction of the fan, the propeller type of fan is preferable. By way of making a line of demarcation, it may be said that the general run of propeller fans become objectionably noisy for most services when working at above ½-in. of water.

Size

The matter of size is not so easy to decide upon. For the propeller type the choice is simple in comparison to the choice of a centrifugal fan. Usually there are several sizes obtainable which will satisfactorily perform the work required. It is necessary to consider both the first cost and the power charges for each of these sizes, which will involve different speeds and different powers. Other things being equal, the noise will be more pronounced as the size increases.

The Tennessee Geological Survey is untiring in its efforts to direct attention to the state's resources, and has recently issued a tastefully gotten up booklet entitled "The Resources of Tennessee in a Nutshell."

One company at Chattanooga will develop 85,000 hp., using the water of the Tennessee River and its tributaries. It is estimated that 75,000 hp. will be developed for farm use alone.

The following table shows the developments at the end of 1910 and the undeveloped power that is still possible in Tennessee:

	Developed Power (1910)		Undeveloped Power	
Rivers	Wheels	Hp.	Minimum	Maximum
Tennessee...	1691	85,548	600,000	1,000,000
Cumberland	397	8,576	25,000	50,000
Green.....	10	140	1,000	2,000
Mississippi..	62	796	1,000	2,000
	2160	95,060	627 000	1,054.000

How Gus Stopped the Leak

By F. Webster

The influence of environment has much to do with the practice of the operating engineer. In districts where competition is keen, and where the highest grade of equipment is installed, the operation of power plants is the most exacting, and every heat unit is guarded.

But out in the open districts, as, for instance, in the coal-mining regions and in the lumber-mill country, the "catch as catch can" methods in all their phases of humor prevail. The fussy-and-particular brand of operating engineer would be entirely out of place where uncovered 4-in. steam pipes 1000 or more feet long are sometimes used, or where 3000-hp. condensing power stations are operated continuously noncondensing. The schemes and practices followed when making repairs in such power stations are many times humorous. One of the troubles in the engine room of the Little Buck mine might be taken as an illustration.

Hot water spurted spitefully at times from a sand hole in the throttle-valve casting, and unwary runners often got "stung," yet all hands around the mine soon learned to dodge hot water as effectively as they did the heels of the mules. Each new year's day brought a new crop of good resolutions for repairing the throttle, but, somehow or other, the repair went undone from year to year, possibly because other duties were as many as the calls of the signal bell from the pit, and because no two men agreed on the same method of repair. However, some things come to those who wait, and when the strike and shutdown came, Gus, the engine operator, made up his mind to fix the leak.

He cleaned out the visible part of the sand hole and built a basin of clay against the opening, binding the clay with an old overall leg around the valve. Then he went to the forge shop and heated up the big babbitt ladle with its 40 lb. or so of alloy that happened to be in it. Happy in the idea of doing a good turn, he began pouring the babbitt into the hole. The first dump spattered, but the hole remained visible. Continuing pouring, the metal disappeared in the hole, which at the end of each pause stood open as ever.

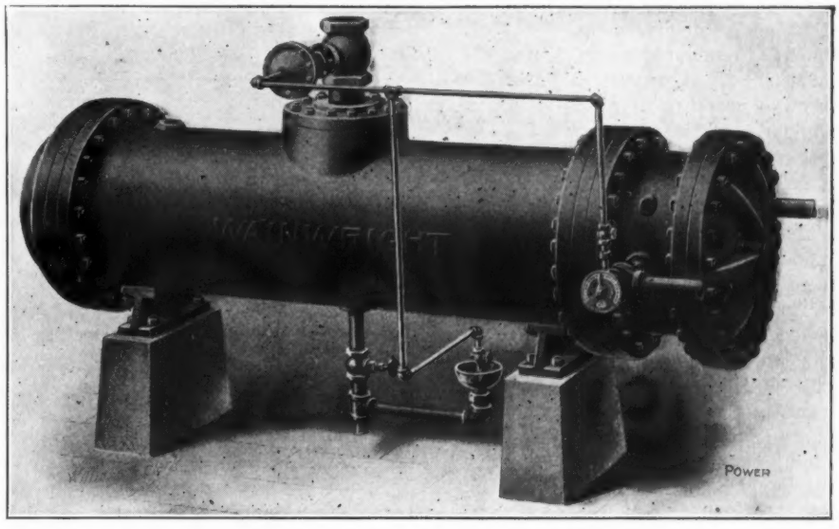
"By tam!" muttered Gus; "I fill him if it takes two pots." With the last drain of dross from the ladle, a puddle of metal held in the clay basin, and Gus rested while it cooled off. Then he removed all the evidences of his work, and filed the babbitt smooth with the valve surface. The hole seemed filled and solid and Gus viewed the job with a grunt of satisfaction.

Six weeks later steam was on and the cage with its first load started from the bottom of the shaft. Everything ran smoothly for a few seconds, when the engine suddenly took the appendicitis. That 40 lb. of babbitt had settled in its interior, and the exhaust was spitting flake babbitt onto the gravel bank back of the engine room. The slide valve shuddered and slammed, and Gus, with his hands on the levers and his eyes riveted on the indicator, stood stiff with fright. Would he be able to stop, or would the engine overwind and wreck the place by dragging the cage over the sheave? It was an anxious minute.

A spurt of hot water came from the sand hole, another tremulous purging of the engine, and it again settled quietly to its work. The car came to the surface and the man aboard stepped from the cage in blissful ignorance of the danger he had so narrowly escaped.

Wainwright Water Heater

The Alberger Condenser Co., 140 Cedar St., New York City, has recently developed a new type of water heater which is designed particularly to fulfill the requirements of the hot-water plant in which the service is intermittent and a constant temperature of the water is desired.



Wainwright Domestic Water Heater

The heater is of the water-tube, instantaneous type, and is made of cast iron throughout with the exception of the heating surface, which consists of the straight, hard-brass Wainwright spiralflow tubing. The tubes are expanded into one tube head and pass through a deep stuffing-box of the condenser type in the other tube head. A spiral coil which agitates the flow of the water binds upon the inner surface of the tubes, so as to secure a very high rate of heat transmission for a given amount of heating surface. This method of accommodating expansion and contraction makes this apparatus applicable for use on either high- or low-pressure steam, for either intermittent or constant service, without any possibility of injurious strains being put upon any part of the apparatus. For installations in which the storage of water is necessary, a storage tank is furnished in connection with the heater.

When a constant temperature of the water is desired, an automatic temperature regulator of an approved manufacture is supplied. The accompanying illustration shows the heater as equipped with the regulator.

List of Water Equivalents

The United States Geological Survey is authority for the water-measurement equivalents given in the following list:

One second-foot equals 40 California miner's inches (law of Mar. 23, 1901).

One second-foot equals 38.4 Colorado miner's inches.

One second-foot equals 40 Arizona miner's inches.

One second-foot equals 7.48 United States gallons per second; equals 448.8 gal. per minute; equals 646,272 gal. for one day.

One second-foot equals about one acre-inch per hour.

One second-foot for one 31-day month equals 61.49 acre-feet.

One million United States gallons equals 3.07 acre-feet.

One California miner's inch equals 0.187 United States gallon per second.

One Colorado miner's inch equals 0.195 United States gallon per second.

One acre-foot equals 325,850 gal.

One cubic foot equals 7.48 gal.

One horsepower equals one second-foot falling 8.8 ft.

The formula

$$\frac{Sec.\text{-}ft. \times fall\ in\ ft.}{11} = net\ hp.$$

may be used to calculate water power on a waterwheel realizing 80 per cent. of the theoretical power.

Gas Power Department

Worth-while gas-engine and producer information treated in a way that can be of practical use

Increasing the Capacity of Large Gas Engines

By F. E. Junge

The problem of increasing the capacity of large gas engines by scavenging with compressed air has recently been attacked with much success by Wilhelm Hellmann, in Hörde, Germany. His investigations were contributed to the transactions of the Society of German Engineers, and the following is a summary of the results:

Theoretical Difference of Output of Ordinary and Scavenged Engines

The following basic values are assumed in the calculated comparison:

Compression space volume, 18 per cent. of piston displacement.

Weight of one cubic meter air and gas.

Coefficient of expansion = 0.00367.

Absolute pressure of gases remaining in cylinder = 1.05 atmospheres.

Absolute pressure of gas-air mixture taken in = 0.90 atmosphere.

Temperature of remanent gases = 500 deg. C.

Temperature of gas and air mixture taken in = 20 deg. C.

The influence of the cylinder walls is neglected in the calculations because the results are thereby not materially affected. With a compression space of 18 per cent. of the piston displacement, the temperature of the mixture of burnt gases remaining in the cylinder and of the new charge of gas and air taken in is 55.5 deg. C. In contact with the fresh mixture, the residual gases in the compression space contract to a volume of 7.67 per cent. of the total, thereby increasing the volume of the new charge of gas and air 18 — 7.67 = 10.33 per cent. Hence the total volume of active charge becomes 100 + 10.33 = 110.33 per cent. of the piston displacement. The engine on which the above calculation is based is a new blast-furnace gas engine built in Germany, in which the relatively small compression space of 13 and 15 per cent. formerly employed has been abandoned in favor of 18 per cent. in order to avoid the high explosion pressures which caused sudden high strains on rods and bearings and reduced the life of the engine.

The influence of the compression space on the temperature of the charge and on its volume, and the effect of scavenging, are shown in the table. The temperature of the mixture increases with the increase of the compression space without scavenging, and the weight of gas and air taken in, and therefore the output, decreases. Moreover, the mixture is considerably impaired by the increasing volume of residual burnt gases and the output of the engine is thereby still further reduced.

In the scavenged engines, for which the absolute pressure of the scavenging air is put at 1.25 atmospheres, after thorough scavenging of the residual gases the gas-air mixture in the cylinder at the end of the suction stroke is equal to the full volume of piston displacement plus clearance space, and at a temperature of 20 deg. C. Compared with the heated mixture of an engine working without scavenging, the volume of active charge in the scavenged engine is 140.2 per cent., which gives an increase in output of 27.3 per cent., using a compression space of 25 per cent. This gain in output is due to the increased volume of the mixture; also to the fact that in a scavenged engine, in spite of lower compression pressures, the same explosion pressure is obtained as with ordinary engines having a compression space of 18 per cent. of the piston displacement.

In contrast to the figures for the ordinary engines, those for the scavenged engines show a constant charge temperature and an increase, with increasing compression space, of the volume of fresh mixture drawn in during the suction stroke. The increase in output over the normal engine is notable. The compression spaces of 30 and 50 per cent. given in the table are of no use in practice because they give a low thermal efficiency owing to the low compression, but a compression space of 25 per cent. gives favorable results.

The calculated gain of output of 27.3 per cent. is due partly to the increased volume and partly to the reduced temperature of the charge, whereby the total weight of active charge is much increased, it being assumed that the available volumes of the scavenged and unscavenged engine, as compared, contained pure mixtures of the same composition. This, of course, is not the case, for the 110 per cent. of the mixture in the ordinary engine is diluted during the suction and compression strokes by the residual burnt gases, which constitute 7.6 per cent. of the stroke volume. Hence the incoming mixture must always have a surplus of air in order to attain good combustion.

Compared with a perfect mixture, the actual mixture in the engine contains less gas and more air than are required. If the 110 per cent. of mixture were not impaired by the residual gases, a more correct ratio of gas and air could be used, and with the same contraction in the volume of burnt gases an increased capacity of 7.67 per cent. would be attained. If these considerations are referred to the scavenged engine, an increased capacity of

$$\frac{7.67 \times 140.2}{110.33} = 9.8 \text{ per cent.}$$

would be realized. The theoretical limit of increased output would therefore be 27.3 + 9.8 = 37.1 per cent.

COMPARISON OF SCAVENGED AND ORDINARY FOUR-STROKE ENGINES

Pressures are absolute, in atmospheres. Temperatures are in Centigrade degrees.

	Nonscavenging			Scavenging		
Clearance volume in per cent. of stroke volume	25	30	50	25	30	50
Compression ratio	5	4.3	3	5	4.3	3
Compression pressure, atmospheres	7.9	6.5	4	7.9	6.5	4
Temperature of residual gases	550	575	650	.....	.....	.....
Aspiration temperature of mixture	20	20	20	20	20	20
Pressure of aspirated mixture	0.9	0.9	0.9	0.9	0.9	0.9
Temperature of total cylinder contents before compression	69.8	79.9	118.5	20	20	20
Volume of concentrated residual gases, per cent. of piston displacement	10.4	12.5	21.2	.....	.....	.....
Charge volume of gas-air mixture, per cent. of piston displacement	114.6	117.5	128.8	125	130	150
Charge volume of gas-air mixture referred to an engine with a compression space of 18 per cent. and a charge temperature of 55.5 deg.	109.7	109.5	108.2	140.2	145.9	168.3
Charge volume of gas-air mixture referred to an unscavenged engine of equal size, per cent	.....	.....	.....	146.1	156.8	200

Prof. Köhler, of Aachen, maintains that the gain through scavenging is even greater. He says an ordinary engine with a clearance of 18 per cent. has a charge volume of 87.3 per cent., while the scavenged engine with 25 per cent. clearance has a charge volume of $0.9 \times 125 = 112.5$ per cent. Assuming that the effective capacity is proportional to the charge volume, the gain of output would be 28.6 per cent. and the total theoretical gain $28.6 + 9.8 = 38.4$ per cent.

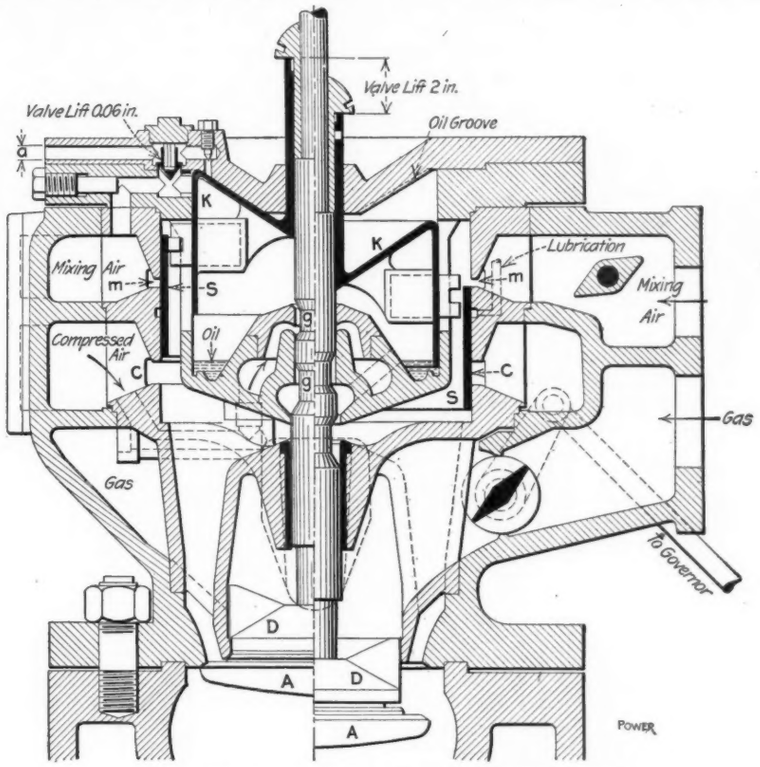


FIG. 1. COMBINED INLET AND SCAVENGING VALVES

DETAILS OF CONSTRUCTION

Compressed scavenging air is conducted to the engine through a special pipe and its entrance into the space behind the inlet valve is governed by a separate slide valve, termed the scavenging valve. This valve can be operated in various ways; one way is by connecting it with an air piston, as shown in Fig. 1, where *K* is the piston. In the left-hand section of the drawing the air piston and scavenging valve are shown in the highest position; in the right-hand section, in the lowest position. On its way down the combined air piston and scavenging valve is mechanically actuated by the valve-gear which opens the main inlet valve. A slight resistance to the downward travel of the slide valve is offered by the vacuum above the piston *K*, which can be adjusted by means of the small air-admission valve *a*.

About 1 per cent. after reaching the dead point the ports for the admission of scavenging air are closed, while the ports for the admission of mixing air remain covered for a while in order to permit, for a fraction of the suction stroke, the entrance of pure gas into the cylinder, where it meets the scavenging air in the compression space. The length of this period of pure gas admission is very important and depends entirely upon the heat value of the gas. It may be extended as far as 30 per cent.

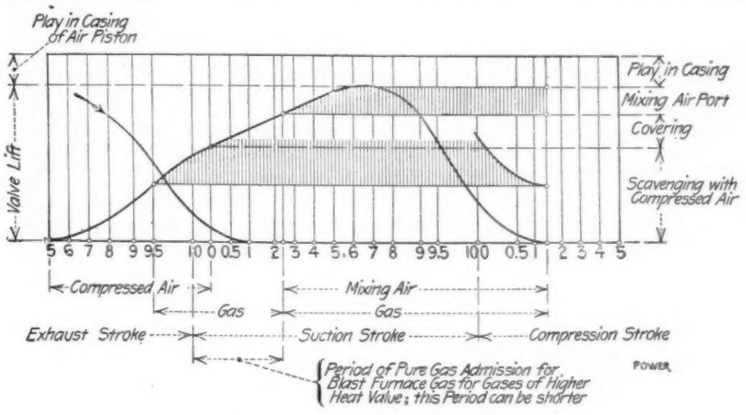

Play in Casing of Air Piston
Valve Lift
Play in Casing
Mixing Air Port
Covering
Scavenging with Compressed Air
5 6 7 8 9 9.5 10 0 0.5 1 2 3 4 5 6 7 8 9 9.5 10 0 0.5 1 2 3 4 5
Compressed Air
Mixing Air
Gas
Gas
Exhaust Stroke
Suction Stroke
Compression Stroke
Period of Pure Gas Admission for Blast Furnace Gas for Gases of Higher Heat Value; this Period can be shorter
POWER

FIG. 2. VALVE-LIFT DIAGRAM

After the valve has reached the dead point. At the close of this period air and gas are admitted together in proper proportions for the rest of the suction stroke.

The various phases of the process are shown in the valve-elevation diagram, Fig. 2. The air piston *K* and scavenging valve *S* being balanced, there is no increase in the load on the valve-actuating gear. The greatest strain is exercised through the closing spring at the highest elevation of the valve. On its way up the combined air piston and scavenging valve first closes the ports *m* for the admission of mixing air and then opens the ports *C* for the admission of scavenging air, which fills the space behind the main inlet valve *A*, so that the expulsion of the burnt gases can begin any time during the exhaust stroke. The upward movement of the main inlet valve *A* and that of the combined air piston and scavenging valve are, of course, accomplished separately.

The seating of the main valve is done by the closing spring, but the lifting of the combined air piston and scavenging valve is done by the compressed scavenging air which enters through a small port formed by grooves *g* around the valve spindle and flows beneath the air piston *K*. A period of 2½ piston strokes is available for lifting the combined air piston and scavenging valve into its highest position. To prevent sticking at the start, the scavenging valve should be lubricated freely and should be actuated from time to time when the engine is standing idle.

In order to keep the compressed air from entering into the gas main in case the gas valve should not work, and to prevent premature ignition and backfiring in the valve chamber, a special gas valve *D* is provided concentric to the main air-inlet valve. The mixing of air and gas, therefore, does not take place until the two media pass the inlet valve on their way into the cylinder, and the admission of the two is closed off at the same time. If for some reason or other it is desired to run the engine without scavenging, the delivery of compressed scavenging air is stopped and the combined air piston and scavenging valve remains in its lowest position, where the ports for the admission of mixing air are open, as will be seen from Fig. 1.

SCAVENGING PUMPS

Unless air under pressure is available from a blower, as is the case in blast-furnace and steel-smelting plants, a special scavenging pump is required, which

is preferably coupled directly to the crankshaft of the gas engine. Fig. 3 shows a double-acting air pump with slide valves, which are best fit for the purpose. The slide valve is superior to the automatic valve not only because it possesses greater operating reliability but because the losses through negative work are lower, provided a pressure of not more than 0.3 of an atmosphere is to be dealt with. The vertical arrangement of cylinder and valve affords easiest lubrication, low losses through friction and inconsiderable wear. Pressure variations of the scavenging air have no influence on the succeeding charging process, because scavenging and charging are independent of each other. High-pressure air for scavenging gives even better results than low pressure, because the expulsion of burnt gases is more thorough.

The pump shown in Fig. 3 serves as an auxiliary for a double-acting tandem gas engine with cylinders 20 in. in diameter and a stroke of 28 in., running at 150 r.p.m. The pump-cylinder diameter is 21 in. and the stroke is 12 in. Under normal conditions, this gas engine has an output of 543 hp. The air required for scavenging must have a volume 1.75 times the compression space, and the latter is 25 per cent. of the stroke volume. Hence the quantity of air needed is 597 cu.ft. per minute and the air pump has to operate at 150 r.p.m. in order to deliver the required amount. The average air-pump diagram shows a compression pressure of 1.25 atmospheres, absolute, and a mean indicated pressure of 0.3 of an atmosphere. The scavenging pump absorbs 11.847 hp., or 2.17 per cent. of the normal indicated power of the gas engine.

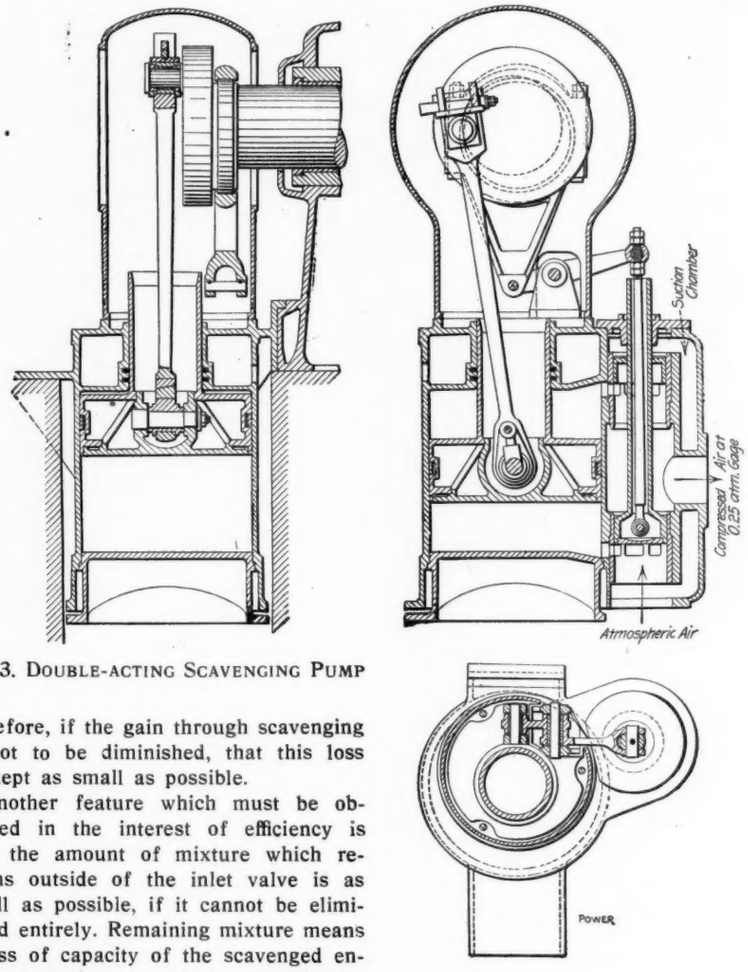


FIG. 3. DOUBLE-ACTING SCAVENGING PUMP

CONDITIONS FOR INCREASING CAPACITY

The scavenging of the compression space, the expulsion of burnt gases, and their replacement by cold air are alone not sufficient to realize a definite increase of output. The scavenging is only a preparatory process and must be supplemented by immediate charging, it being essential that to the air in the compression space is added a corresponding quantity of gas without additional losses. In order to attain this the pressure of the scavenging air must be in the neighborhood of 1.25 atmospheres, absolute, and the cross-sections of the passages in the inlet casing must be of ample proportions to avoid throttling. There is no throttling in the pipes, because the scavenging and the mixing air are conducted separately. Hence, the scavenging air can expel the burnt gases with undiminished force, beginning during the middle of the exhaust stroke when the valve opening has reached its maximum.

It is essential to make the space between the inlet valve and the scavenging valve as small as possible, so that the air in the inlet-valve chamber assumes the suction pressure immediately after the beginning of the suction stroke, thereby assisting the influx of gas into the cylinder. Hellmann admits that in an ordinary unscavenged engine with a compression space of 18 per cent., the loss due to ineffective expansion is 1.9 per cent., while in a scavenged engine with a compression space of 25 per cent. plus 5 per cent. in the inlet chamber, the loss is 3.2 per cent., assuming expansion in both cases from 1.05 to 0.95 atmospheres absolute. It is essential, therefore, if the gain through scavenging is not to be diminished, that this loss be kept as small as possible.

Another feature which must be observed in the interest of efficiency is that the amount of mixture which remains outside of the inlet valve is as small as possible, if it cannot be eliminated entirely. Remaining mixture means a loss of capacity of the scavenged engine proportionate to the amount of mixture; instead of this mixture, which is scavenged into the exhaust pipe, a corresponding volume of air is taken into the cylinder at the beginning of the suction stroke. Hence, when the scavenging process is over no additional air should be admitted, but gas should be introduced in sufficient quantities to make a perfect mixture. The combination valve shown in Fig. 1 is so constructed that no mixture remains outside of the inlet after the latter is closed, because the mixing of the two media is performed while they pass into the cylinder.

A further essential condition for successful scavenging is that the period of gas influx be as long and the cross-section of gas passage as large as possible. This is especially important when driving dynamos, but for slow-speed gas blowers the period of gas intake may be shorter. Gas admission should begin 5 per cent. before the piston starts on its suction stroke and should not end until 10 or 15 per cent. after the suction stroke is completed. The aspirating action of the compressed air flowing in together with the construction of the inlet, which acts like an injector, causes the acceleration of the gas to begin as early as 5 per cent. before the dead center. This effect may be augmented by a suitable layout of the exhaust piping, though the latter depends largely on local conditions.

When operating on blast-furnace gas it is advisable to continue the period of pure gas admission during 25 per cent. of the suction stroke; then gas and air enter into the cylinder in the usual manner. If these precautions are taken the cylinder at the beginning of the compression stroke contains an undiluted charge, which gives perfect combustion, as shown by the indicator cards. But it must be emphasized again that a notable increase of output by scavenging can be realized only when the losses of the process are reduced as far as possible.

Lack of regard for detail construction is responsible for the fact that the earlier types of scavenging engines have not met with complete success. They had no separate organs for controlling the admission of scavenging air. The inlet valve of the gas engine was the pressure valve of the pump. The space between this valve and the piston of the air pump was very large and this space was still further increased by the addition of the compression space.

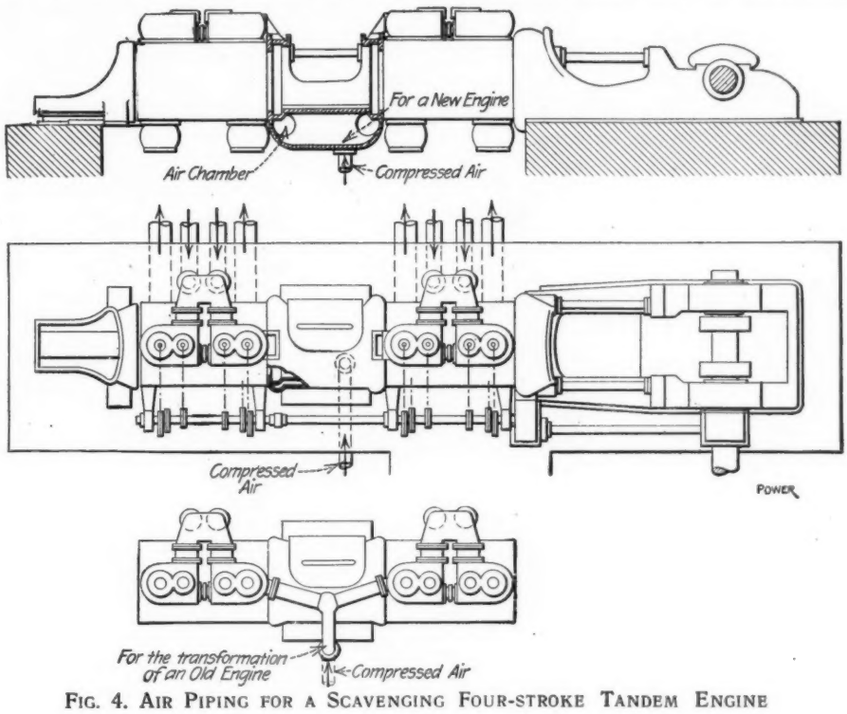


FIG. 4. AIR PIPING FOR A SCAVENGING FOUR-STROKE TANDEM ENGINE

At the beginning of the suction stroke these large spaces were filled with compressed air which expanded during a considerable portion of the suction stroke and prevented the aspiration of gas in due time. Moreover, at the end of the suction stroke of the pump the air in these spaces was at a pressure below the atmospheric, which was transmitted to the charge in the working cylinder; hence, this cylinder was not filled up to its full capacity and the gain through scavenging was accordingly reduced. Hellmann estimates the loss through expansion in the old scavenging engine at 8.7 per cent., and the loss due to mixture remaining in front of the inlet valve at 8 per cent., making a total loss of 16.7 per cent. On this basis the superiority of the new over the old scavenging engine would be 16.7 — 3.2 = 13.5 per cent.

THERMAL CONSIDERATIONS

The indicator diagrams in Fig. 5 are obtained with a clearance volume of 25 per cent. and show a mean effective pressure of 7.2 and 7.3 atmospheres, with compression pressures of 7.5 and explosion pressures of 19 and 19.5 atmospheres, respectively. With blast-furnace gas of the usual heat value, mean effective pressures of 693 normal and 790 maximum are obtained. The corresponding mechanical efficiency is 82 per cent. and 84.5 per cent. respectively. While theoretically there is a difference of 5 per cent. in the thermal efficiency in favor of the smaller compression space (18 per cent.), this theoretical reduction is more than neutralized by the superior ignition and combustion of the chemically favorable and pure mixture in the larger compression space (25 per cent.), a fact which is established by the higher thermal efficiency of large two-stroke-cycle engines. As will be seen from the indicator cards, the explosion pressures do not rise above the normal (about 20 atmospheres) which has been found good practice for large work, while the output is considerably increased.

The operating reliability of scavenged engines is by no means less than that of ordinary engines. On the contrary, owing to the cooler charge, the temperature of the cycle is reduced during the whole of the suction stroke. Again, the temperature is reduced during the whole of the compression stroke, owing to the lower compression pressures used. Moreover, about 50 per cent. after the beginning of the exhaust stroke, scavenging sets in, whereby the mean temperature of the exhaust period is also reduced. In addition there is the favorable expansion due to perfect combustion, which gives an expansion curve approaching the adiabatic.

There is only one phase of the process when temperatures rise above the ordinary, namely, during explosion. For the rest of the time the cyclic temperatures are lower. Therefore the life of the engine, and especially the durability of the flanged exhaust socket, which always shows an inclination to crack as a result of overheating, is increased.

With equal heat value per unit of charge volume, the heat of the charge in large gas engines increases as the cube of the cylinder diameter, while the heat-conducting wall surfaces increase only in the square, compared to a smaller engine. Also the thickness of the walls increases and so does the temperature, in spite of water cooling. Perfect scavenging is apt to equalize the increasing internal temperature of large gas engines, to prevent burnt particles of the lubricant from causing preignition and thereby to increase the operating reliability of the plant.

EXHAUST PIPING

It is known that the design of the exhaust piping has a great deal to do with the efficiency of gas-power plants, because all obstacles causing eddy currents and back pressure reduce the output. The stop diagram in Fig. 6 shows the effect of poor exhaust-pipe arrangement and that in Fig. 7, which was obtained under the same general conditions, shows the improvements realized by facilitating the conduction of the waste gases into the atmosphere. The first two diagrams, of which the one is obtained without and the other with the injection of water into the exhaust, show the resistance which the outrushing gases encounter during 15 or 20 per cent. of the exhaust stroke. They also show that at some 20 per cent. before the end of that stroke the vac-

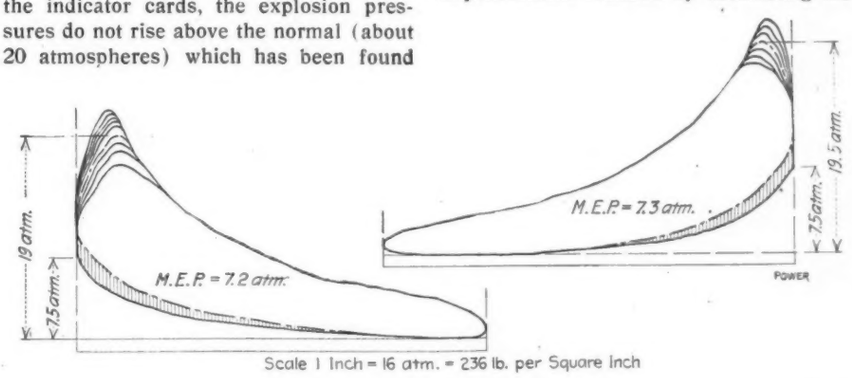
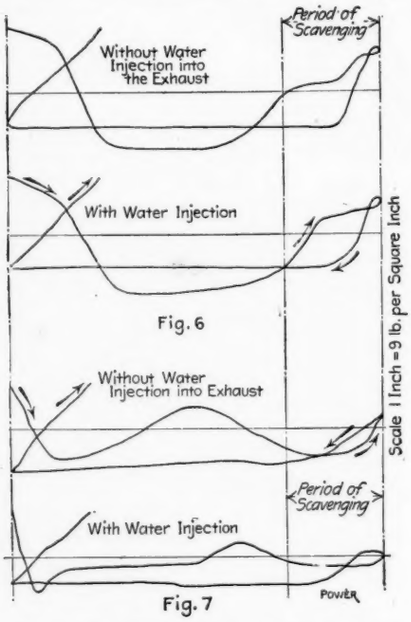


FIG. 5. DIAGRAMS FROM SCAVENGING ENGINE WITH COMPRESSION RATIO OF 5

ıum is turned into a back pressure of 0.1 to 0.16 of an atmosphere, which prevails during the period of scavenging and makes the latter process even temporarily inefficient.

Another result of the back pressure in the latter part of the exhaust stroke is that the suction line does not begin until 6 to 9 per cent. beyond the dead center, whereby a complete filling of the cylinder



STOP DIAGRAMS, SHOWING EFFECTS OF DIFERENT EXHAUST PIPING

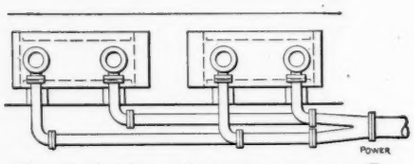


FIG. 8. A GOOD ARRANGEMENT OF EXHAUST PIPING

with active charge is made impossible. With the improved layout of the exhaust piping there is much less resistance in the early part of the exhaust stroke and atmospheric pressure prevails during the scavenging period, whereby scavenging and charging are made more effective.

Experiments have shown that it is advantageous to lay out the exhaust piping for both sides of the cylinder in the same manner, and that curves or elbows which may cause eddying must be avoided as much as possible in order to secure an even flow. Fig. 8 shows how to lay out the exhaust piping for large gas engines. With this arrangement the exhaust of one combustion chamber has no clogging influence on the scavenging which goes on in another combustion chamber at the same time.

Summing up, the Hellmann system affords a notable increase of capacity at low additional cost. The mean temperatures and pressures are not higher than they are ordinarily, while the engine output is increased 25 per cent. and more. Preignition is avoided and heat consumption improved. By choosing a smaller type of engine the bulk of plant is reduced and the ratio of plant output and initial cost becomes more favorable. By shutting off the scavenging air the engine continues to run in the ordinary manner. Any existing type of large gas engine may be thus improved.

LETTER

Effect of Temperature on Crude Oil

BY C. E. HOWELL

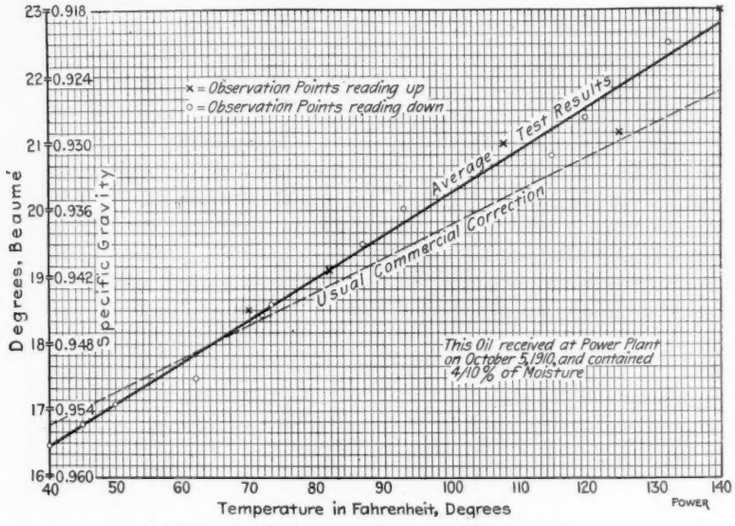
In the Nov. 14, 1911, issue of POWER there was a short article by a Mr. Lyons on the effect of temperature on the specific gravity of crude fuel oil, accompanied by a curve of the results of tests made to determine this effect. The article attracted the attention of the writer because he had made a similar test about a year previously.

Upon unearthing the curve made by himself and plotting the results on the chart in POWER, it was found that the points of both the rising temperature and the falling temperature tests coincided exactly with those in the POWER chart. Furthermore, the percentage of moisture in the oil used was the same, $\frac{1}{10}$ per cent. The writer of the article in the Nov. 14 issue gave Oakland, Calif., as his address, but the test by the present writer was made in Los Angeles. Investigation fails to indicate that any copies of the result of the Los Angeles test have ever been distributed. The writer takes it that it is merely a very unusual sort of coincidence. However, it may not be amiss to explain the present chart.

Primarily, the data were taken to check the well known purchasing method of deducting or adding one degree Baumé for every 20 deg. F. above or below 60 deg. F., respectively, to reduce the specific gravity of the oil purchased to what it would be at 60 deg. F. The chart indicates that 1¼ deg. Baumé should be deducted or added, as the case may be, for each 20 deg. F. difference in temperature. The chart has since been used in obtaining the approximate gravity of fuel oils at different temperatures while heating them for firing purposes. By also giving the specific gravity with water as the basis of comparison, the correction for expansion may be found.

The commercial correction for the effect of temperature on the bulk of crude oil is 1 per cent. for each 20 Fahrenheit degrees of difference in temperature. Referring to the correct curve, however, one finds that the specific gravity of this oil at 60 deg. F. is 0.9495, while at 80 deg. it is 0.9420. The percentage of expansion in raising the temperature from 60 deg. to 80 deg. F. is

$$\frac{100 \times 0.9495}{0.942} - 100 = 0.8 \text{ or, } 1 \text{ per cent.}$$



EFFECT OF TEMPERATURE ON CRUDE OIL

and the oil companies therefore get the worst of it on the expansion correction for change in temperature when using the old allowance of 1 per cent.

A Thirty Day Run

A British firm is building two sets of eight-cylinder, four-cycle Diesel oil engines of 2500 combined horsepower for the "Jutlandia," which was recently launched. One of these engines was rigged up on the "bench" and attached to a dynamo. It was then given a 30-day continuous night and day nonstop run, 28 days of which were at full power, one at three-quarters and one at half power, burning Scotch shale fuel oil of a specific gravity of 0.85. During the 30 days' run the engine was never stopped or adjusted in any way, no trouble whatever was had, and it ran as steadily on the last day as on the first.

Electrical Department

Conducted to be of service to the men in charge of electrical equipment in the power house

Westinghouse Belt Driven Alternators

The accompanying engravings illustrate the construction of the latest form of Westinghouse belt-driven alternating-current generator designed particularly for operation in industrial plants and small central stations. Having in mind the usual conditions of operation in such plants, special efforts were made to obtain as simple and rugged construction as possible.

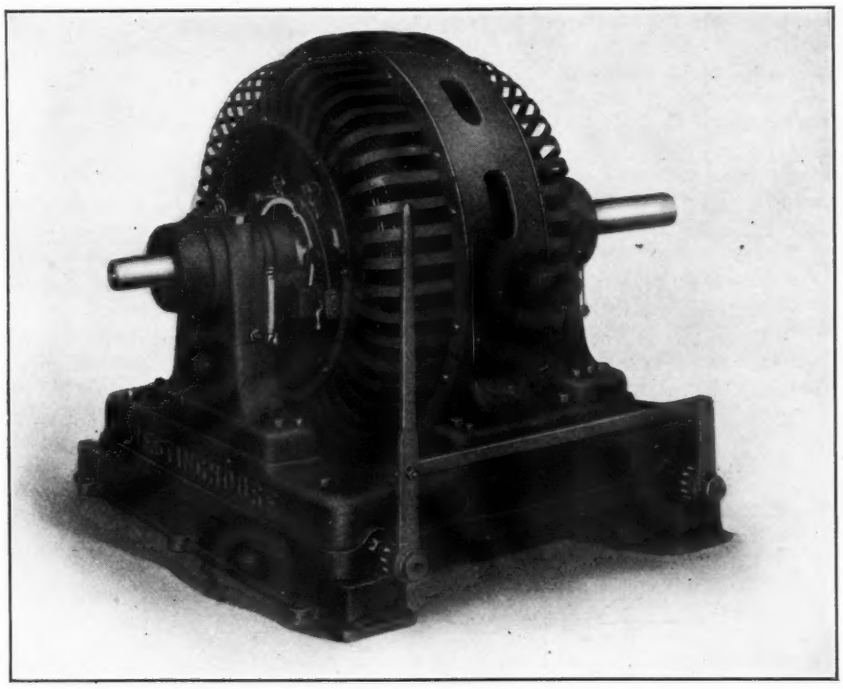


FIG. 1. WESTINGHOUSE BELT-DRIVEN ALTERNATOR

The field magnet is the revolving member and it consists of a laminated steel spider (Fig. 2) carrying laminated magnet poles which are held on the spider by dovetails. The dovetail slots are stamped in the disks of the spider and the corresponding dovetail is stamped in the steel sheets forming the magnet pole. The dovetail on the pole is smaller than the slot in the spider and is secured in the slot by means of two taper keys extending the whole length of the slot. Fig. 3 shows the construction of a magnet pole. The rectangular slots near the edge of the pole face are provided to take the bars of a squirrel-cage winding, arranged as shown in Fig. 4.

The function of the squirrel cage around the magnet pole faces is to "damp out" any tendency to "hunt" when operating in parallel with other machines. The squirrel cage also partly neutralizes the armature reaction of single-phase machines and thereby improves the regulation. The construction of the cage is so clearly shown in the picture that a detailed description is unnecessary. It is supplied only on machines of 75 kilovolt-amperes and larger.

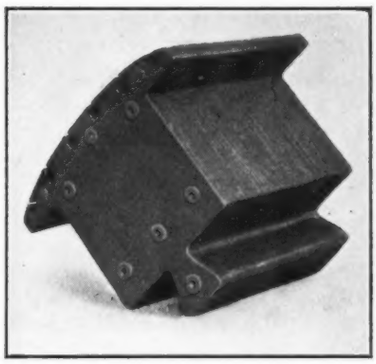


FIG. 3. FIELD-MAGNET POLE

Fig. 5 shows a field-magnet spider with the poles and coils in place. The coils are of the usual form-wound type, completely insulated before being put on the cores. It will be evident on close inspection that two retaining straps are

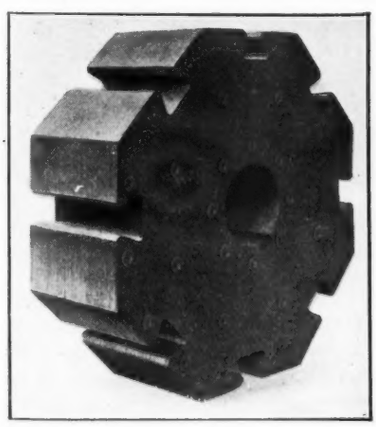


FIG. 2. FIELD-MAGNET SPIDER

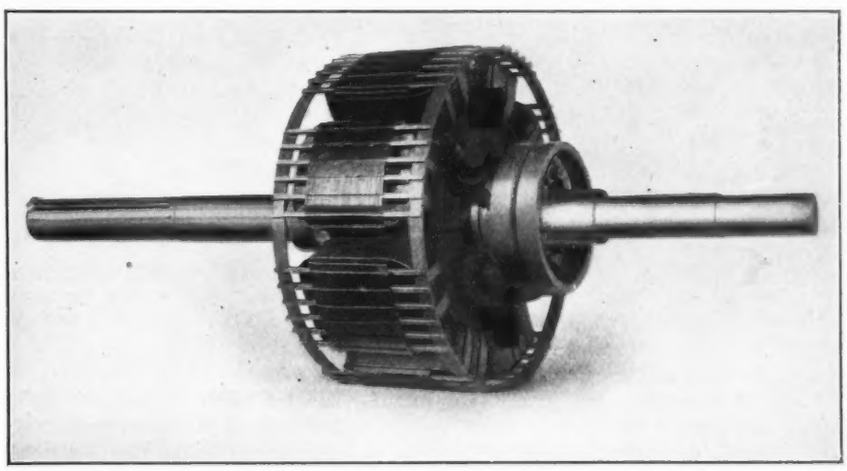


FIG. 4. COMPLETE FIELD MAGNET, WITH SHAFT AND SLIP RINGS

applied to the end of each magnet coil in such a way as to restrain the projecting portions from spreading radially under the influence of centrifugal force. The polar extensions protect those parts of the coil lying parallel with the shaft.

The armature core is of the usual laminated type, provided with entirely open slots and mounted in a substantial cast-iron housing, as shown in Fig. 6. The armature coils are form-wound and individually insulated; they are held in the core slots by fiber wedges driven in small slots in the flanks of the core teeth.

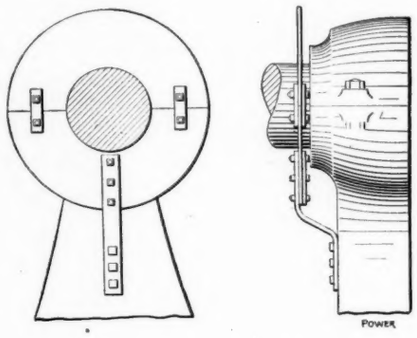
In sizes above 75 kilovolt-amperes the shaft is supported by pedestals, as illustrated in Fig. 1; in smaller sizes, circular journal brackets are bolted to the stator housing as in the case of small direct-current machines. These generators are built by the Westinghouse Electric & Manufacturing Co. in sizes ranging from 30 to 200 kilovolt-amperes, for 60 cycles frequency and for all standard alternating-current voltages from 240 up to 2400.

LETTERS

Keeping Oil off the Commutator

I once had charge of an electric generator which was directly driven by its engine, the armature being mounted on a crankshaft extension, between the engine flywheel and the outboard bearing. Continuous trouble was experienced by oil being thrown out of the bearing onto the brush gear and commutator, due to the shaft whipping slightly in the out-end bearing block. Drops of oil would be thrown from the top of the brass and no matter how finely the box was adjusted no cure for the trouble was effected. The maker's representative saw the job and admitted that it must be some balancing fault in the crankshaft (running at 250 r.p.m.) which caused the oil to be jerked out so.

He left without suggesting a remedy and as a last resort I fixed up the guard or shield shown in the accompanying drawing. The shield was made out of $\frac{1}{16}$-in. sheet brass held by braces $\frac{3}{16}$x1 in. and $\frac{3}{16}$x1½ in., of mild steel. The hole in the brass disk was made ⅛ in. larger in diameter than the shaft, which allowed $\frac{1}{16}$-in. clearance all around, and the disk was split horizontally. The drawing shows the method of mounting on the brass plate and no further trouble has been experienced from this cause.



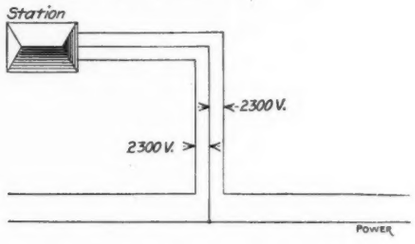
MR. BLACK'S OIL GUARD

J. E. BLACK.

Manchester, Eng.

An Improvised Three-Phase System

I was called on a short time ago to install a 2300-volt three-phase generator and switchboard to take the place of an 1100-volt single-phase machine that had burned out. The owners did not have the time, nor the desire, to go to the expense of wiring the town for three-phase distribution and the new machine was not large enough to carry the load running single-phase.



IMPROVISED THREE-PHASE SYSTEM

I opened one side of the old main line near the center of distribution and connected it to the station as shown in the diagram. The load is as nicely balanced as any that I have seen. As most of the transformers were of modern types with 1100- or 2200-volt primary windings they did not give much trouble. All 1100-volt transformers that were worth saving were connected in series and used.

D. W. SCARBOROUGH.

Richmond, Tex.

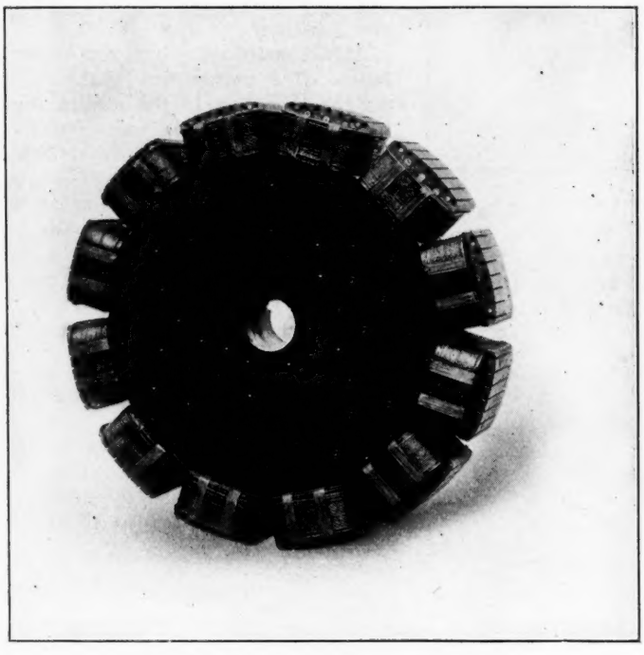


FIG. 5. FIELD MAGNET, WITH COILS

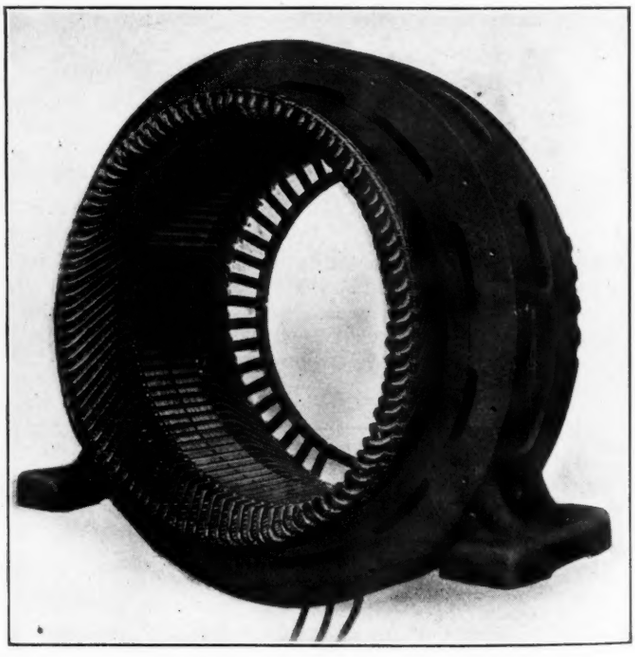


FIG. 6. ARMATURE COMPLETE IN HOUSING

Heating and Ventilation

Considered as power-plant problems. Layout and operation of systems and apparatus

Heating and Ventilating System of the Ritz-Carlton Hotel

By Charles A. Fuller

The heating and ventilating system designed for the addition to the Ritz-Carlton Hotel, at Madison Ave. and Forty-sixth St., New York City, has several features which may perhaps be of interest to engineers in this line of work. Because of the unusual arrangement of the building, it was necessary to divide the ventilating into four separate systems as follows: ball room and banquet hall, private dining room, service floors and toilet-room exhausts.

The system for the ball room and banquet hall is the largest of the four, handling about 50,000 cu.ft. of air per minute. These rooms are each 50 ft. wide and 90 ft. long. The banquet room is located below the street level and has a ceiling height of 20 ft. The ball room is located directly above the banquet hall and has a ceiling height of 30 ft. Above the ball room is located the blower room, containing all the ventilating apparatus for this system.

The air in entering first passes through a tempering coil, then through an air washer to the intake of the supply fan. From the supply fan, it passes through a reversing damper and the reheater to the distributing ducts. The tempering coil consists of two banks of American Radiator vento sections 40 in. high. Each bank is divided into four groups, two groups high and two groups wide, placed back to back. Each group contains 16 sections, making a total of 1376 sq.ft. of radiating surface.

The steam supply to the first bank is controlled by a thermostat placed in the cold-air inlet, and the steam supply to the second bank by a thermostat placed between the second bank and the air washer. The first thermostat is adjusted to close the steam supply to the first bank when the temperature of the entering air is above 25 deg.; the second is adjusted to close the steam supply to the second bank when the air passing through the tempering coil is above 50 deg. This arrangement keeps the temperature of the air entering the air washer at 50 deg., regardless of variations in the outside temperature, so long as it is below 50 deg. This is necessary in order to maintain a constant humidity in the rooms, which is a very desirable feature.

The reheater consists of three banks of American Radiator vento sections 60 in. high. Each bank is divided into two groups placed back to back, and each group contains 16 sections, making a total of 1536 sq.ft. of radiating surface. The main steam supply to the reheater is equipped with a diaphragm valve controlled by a thermostat placed in the hot-air duct leading from the heater.

A bypass is provided below the reheater. The hot air from the reheater and the tempered air from the bypass then pass through thermostatically controlled mixing dampers which operate so as to produce the required temperature in the room supplied. With this system, each room must have a separate duct and a set of mixing dampers, which insures that the required amount of air is being constantly delivered to each room regardless of outside temperature changes.

After determining the heat losses from the various rooms, it was found that the required temperature of the air from the reheater for extreme conditions was approximately 160 deg. This temperature could be obtained with the tempering coil as described and a reheater composed of two banks and maintaining a velocity through the heater of 1200 ft. per minute, requiring a free area of about 42 sq.ft. As the available head room was only 7 ft., the necessary width made this arrangement practically prohibitive.

It was then decided to put in three banks of heaters, which would make the resulting temperature 120 deg. with a velocity of 1200 ft. per minute. Then by passing only 75 per cent. of the air through the heater and 25 per cent. through the bypass in the extreme conditions, the resulting temperature directly beyond the mixing dampers would be 100 deg., and by so doing the width of the reheater was cut down about 25 per cent. This arrangement, however, does not reduce the amount of heating surface necessary.

Means for cooling the air in summer time, in addition to the cooling effect of the air washer, is also provided by direct-expansion cooling coils placed in the spray chamber of the air washer. A constant spray of water is kept flowing over the coils, which prevents the formation of frost on the outside of the pipes. The humidity is controlled by cooling the air to a definite point below the outside temperature, which also lowers the temperature below the dew point and some of the moisture is condensed. This condensed moisture is removed when passing through the eliminator. The temperature of the air in passing through the distributing ducts and into the rooms is raised due to the warmer outside temperature and, consequently, the humidity is lowered to the desired point.

Each of the rooms supplied by this system has a series of registers at the floor level and also at the ceiling level. During the winter time, when warm air is supplied, this air enters through the registers located at the ceiling and the floor registers serve as the exhaust registers. In summer, however, when cold air is supplied, the floor registers serve as the supply and the exhaust is taken out through the ceiling.

With this arrangement, the best possible ventilation is secured. In winter time, the air enters the room at a higher temperature than the room and is cooled after entering. This causes a natural flow of air downward, and for this reason, the proper point to remove the foul air is at the floor level.

In summer, the air enters the room at a lower temperature and is heated after entering. This produces an upward flow of air and for the same reason the proper point to remove the foul air is at the ceiling; this is particularly true of rooms with very high ceilings where air cooling is desired. If the cold air were introduced at the ceiling, all the refrigerating effect would be lost by the time the air had reached the breathing line.

This reversing of the direction of flow is produced by means of a reversing damper shown in Fig. 1. In the position shown on the drawing, the damper is set for cold-weather conditions or for delivering warm air to the rooms through the ceiling registers. The floor registers are connected to a system of vertical flues which combine into one main duct passing up through the floor to the blower room at the point *A* and thence through the opposite compartment of the reversing damper to the exhaust fan.

To deliver cold air from the air washer to the rooms through the floor registers, the damper is set at right angles to the position shown. The fresh air then passes

through the washer and supply fan and then the reversing damper to the main duct leading to the registers. The exhaust fan draws the air from the ceiling registers back in the opposite direction through the distributing ducts to the exhaust fan.

A recirculating duct is also provided to recirculate part of the air in summer when the ventilating conditions are not severe. This cuts down the amount of refrigeration necessary for cooling the air, as the air being drawn out by the exhaust fan contains considerably less moisture than the outside air and, consequently, it is not necessary to condense as much moisture by this arrangement.

This system is designed to change the air in each room once every six minutes, which is adequate ventilation for all rooms when entirely filled. For ordinary use, however, at least one-half of the air can be circulated and still be well within the required limits of good ventilation.

All cold-air supply ducts are provided with a covering of cork board 1½ in. thick to prevent the collection of moisture on the outside and also loss of refrigeration.

Two small coat rooms are supplied from this system, the rooms being so small in comparison with the others that it was impractical to run separate ducts to them and provide each with its own set of mixing dampers for controlling the temperature. Branches were taken to them from the supply and exhaust duct to the entrance hall.

The coat rooms have a larger proportional amount of exposed wall and glass surface and therefore the temperature of the entering air would not be high enough to maintain their temperature at 70 deg. To compensate for this, an auxiliary indirect heater was placed in the supply duct to each room and the steam supply to the heater controlled by a thermostat placed in the room.

To determine the amount of radiating surface to install in these ducts, the following computation was used: The total heat loss from the entrance hall was first determined in the usual manner. The air change in this room was six times per hour. If the resulting temperature in the room is 70 deg., then

$$\frac{B.t.u.\ loss \times 55}{Cubic\ contents \times 6} = T - 70$$

where T is the temperature at which the air should be introduced. Applying this formula, T was found to be 88 deg. in zero weather.

The same formula was applied to the coat rooms, except 10 changes of air were used instead of six, which gave a required temperature of 108 deg. for the entering air. It was then necessary to put in radiation enough in the duct at the point where it entered the room to raise the temperature from 88 to 108 deg., or through a range of 20 deg. Referring to the tables given in the American Radiator

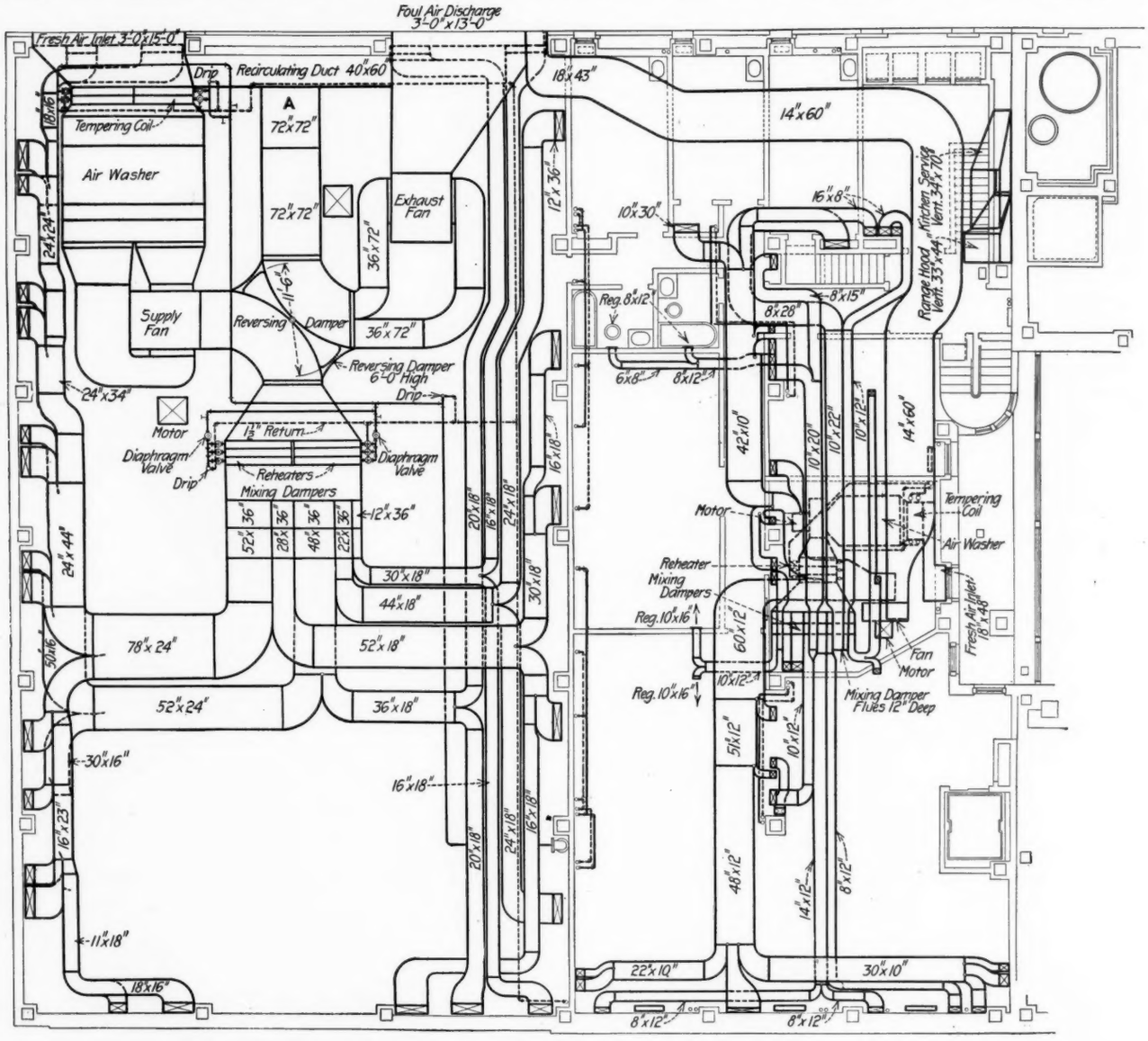


FIG. 1. LAYOUT OF HEATING AND VENTILATING SYSTEM FOR RITZ-CARLTON ADDITION

data book, it was found that with a velocity of 600 ft. per minute through the heater, which was the velocity of the air in the flue, one bank with sections spaced on 5⅜-in. centers would give the required rise in temperature. Then dividing the total quantity of air by 600 gave the required amount of full area. The given free area through each section divided into this then gave the number of sections and the amount of surface necessary.

The toilet rooms are vented by means of one No. 9 multi-vane fan placed on the top floor and discharging to the atmosphere through a standard ventilator constructed of 24 oz. copper. Each toilet room has a register and duct leading to the main vent flue, the duct passing into the flue and up for a distance of about 3 ft.; in the vertical portion a controlling damper is placed. The top of each duct is covered with a tempering cap to prevent rubbish from falling into the duct

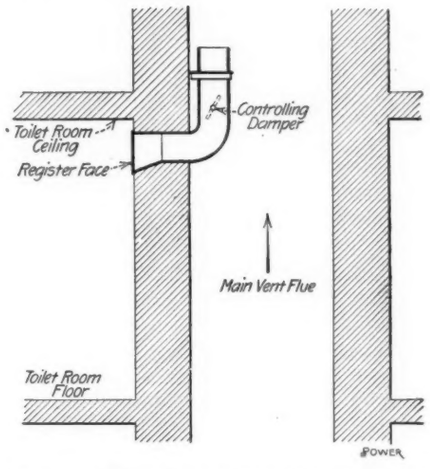


FIG. 2. SECTION THROUGH TOILET ROOM AND MAIN VENT FLUE

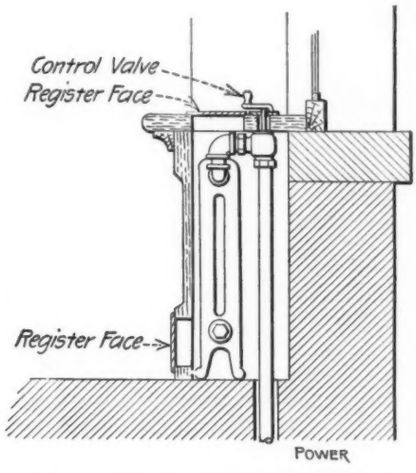


FIG. 3. INCLOSED RADIATOR UNDER WINDOW

during construction; after the work is completed, the caps are removed.

There are four of these main vent flues in various portions of the building, all of which lead from the first to the twelfth floor, where they combine into one main duct running in the hung ceiling over the hall and thence to the exhaust fan. In all toilet rooms throughout the buildings the air is changed once every four minutes. No fresh-air ducts are taken to the toilet rooms, but each room, whether it has outside exposure or not, has an inclosed direct radiator.

The circulation of steam throughout the building is produced by means of a vacuum system. Thermostatic traps containing a vaporizing fluid are used on the return of each radiator and heater and the inlet to each radiator is equipped with a graduated supply valve.

The original building was heated by a vapor system without the use of a vacuum pump. The main return was vented to the atmosphere in the boiler room and then passed through a loop seal to the feed-water heater. With the addition, however, two 8 and 14 by 16-in. vacuum pumps were installed and cross-connected, so that either or both pumps could work on either part of the building. The return from the new building was sealed with a loop and vapor pipe taken out from the return in the same manner as the old building. These pipes were provided with a shut-off valve and the connections to the pumps were taken from the bottom of the seal. With this arrangement, the vacuum pump could be cut out from either building and the system operated on the vapor principle.

The radiators in all the rooms are inclosed in recesses under the windows with registers at the bottom and top of the inclosures. The steam supply enters the radiators at the top through the graduated supply valve, and is so arranged as to bring the valve close up against the top of the inclosure. The valve stem extends through and the lever handle with the indicator is placed on the outside, where it is very accessible for the occupant to manipulate and obtain the amount of heat desired by hand regulation. A cross-section of this inclosure, illustrating connections to the radiator, is shown in Fig. 3. The forced-draft system of the plant will be described in a later issue.

The heating and ventilating system was designed by Clark, MacMullen & Riley, consulting engineers of New York City, and is being installed by the Cleghorn Co., of Boston, Mass.

The successful engineer is the man who, in addition to solving the purely mechanical problems, can make the dollar go farthest and draw out the best efforts of his workmen. Knowledge and study of the human factor bulk large in executive success.—*General Electric Review.*

An Isolated Plant Would Save $24,000

BY IRA N. EVANS

The paper read by Davis S. Boyden before the American Society of Heating and Ventilating Engineers and published in POWER of Feb. 27, is very clear and complete where the cost of the heating is concerned, but a little foggy when it comes to the contract rates for electric service in the stores of J. M. & Co., of Boston. However, by multiplying the area 637,800 sq.ft. by the rate $0.08436 given in Mr. Boyden's article and the cubic contents 8,391,000 cu.ft. by the rate $0.00641, the contract price for current is obtained as $53,805 and $53,786.31, respectively, for the year. Dividing $53,800 by the number of kilowatt-hours used per year gives 1.97c., or a trifle under 2c. per kilowatt-hour.

The reports to the public service commission of the New York district show that the cost of current, with all overhead and distribution charges included, for the New York Edison Co. is 3.16c. per kilowatt-hour. The cost of production for the Boston Edison must be considerably lower for the company to make a profit on a 2c. rate.

The manner in which the live steam for heating was manipulated during the test described is open to question, although the records as given check up with absolute accuracy. It should be noticed that the shortest day in the year, Dec. 20, was taken for the test. Department stores are generally lighted to a greater extent during the day than other buildings, but if daylight affected the load at all it would be boosted to the maximum on the winter solstice. The weather conditions and outside temperatures were a fair average for the heating season.

The method of conducting the heating test is open to criticism for the following reasons:

It is stated that the heating system of the Avon St. store, aggregating 11,000 sq.ft., or nearly 50 per cent. of the direct radiation, had been shut off from 6 p.m. to 12:30 a.m., when the test started. It is also stated that it was necessary to open all the air valves.

This indicates, and an analysis clearly shows, that the heating is shut off nights when the temperature is above 45 deg. outside. The writer has records which clearly show that nearly the same coal will be required, if the heating plant in a large building is shut down for 6 hr., to raise the temperature again, as would have been consumed if the heating were maintained continuously. This means that the coal would have to be consumed at double the rate for the shorter period in raising the temperature in the buildings. If the heating system of the Avon Street Building had been

operated for the previous 6 hr., about 4000 lb. less steam would have been used per hour from 1 to 7 a.m. Thus if the plant had been continuously operated, furnishing exhaust steam, the amount of coal for live-steam heating would have been materially reduced.

The glass and wall areas and transmission of heat of the radiation balance. The air supply when turned on at 8 a.m. shows practically the same steam requirements with the direct radiation included, as before the fans were started. See Fig. 3 of Mr. Boyden's article. In computation the following quantities will be used:

55,000 cu.ft. of air raised 1 deg. will require 1 lb. of steam.

1.7 B.t.u. will be used as the transmission for the direct surface per square foot per hour per degree difference.

222 deg. will be used as the temperature of the radiation at 3 lb. pressure.

24,000 sq.ft. is the number of square feet of direct radiation.

108,000 cu.ft. per minute is the amount of fresh air used for ventilation. It will be calculated to be raised to 70 deg. from the outside temperature.

The direct radiation will require

$$24{,}000 \times 1.7 \times (222 - 70) = 6{,}201{,}600$$

B.t.u. per hour

To heat the air for ventilation will require

$$\frac{108{,}000}{55{,}000} \times 60 = 118 \text{ lb.}$$

of steam per hour per degree rise in temperature.

The evaporation will be taken as 9 lb. per pound of fuel. From the Mar. 12 issue of POWER, Table 2 will be used to obtain the number of hours' operation for different periods of outside temperatures. The fans will be considered in operation during week days from 8 until 6 p.m., as shown by Mr. Boyden's article, or 1545 hr., which is equivalent to 154.5 days of 10 hr. per heating season.

Taking the latent heat as 964 B.t.u. and dividing, the average steam for heating the direct surface will be

$$\frac{6{,}201{,}600}{964} = 6430 \text{ lb.}$$

This will be the load on the direct radiation in average winter weather for heating during all periods, excepting those ranging between 50 and 60 deg. outside temperature. From the table the number of hours is found to be 3761.

There will be more steam used for periods 0-10 deg., 10-20 deg. and 20-30 deg. than for the average weather. From actual operating conditions on a plant the writer has the steam requirements as follows:

Outside Temp., Deg. F.	Lb. of Steam	Ratio	Per Cent.
0–10	16,500	$\frac{165}{112}$	147
10–20	14,760	$\frac{1476}{1120}$	132
20–30	12,960	$\frac{1296}{1120}$	116
30–40	11,200	$\frac{112}{112}$	100

The number of hours will be taken from Table 2 of Mar. 12 for those periods and added to the average steam consumption, 6430 lb. per hour, as 100 per cent.

Outside, Temp., Deg. F.	Hours	Excess, Per Cent.	Excess Steam, Lb. per Hr.	Total Excess Steam, Lb.
0–10	66	47	3020	199,320
10–20	264	32	2058	543,112
20–30	829	16	1028	852,212
				1,594,644

$$6430 \times 3761 = 24{,}183{,}230$$

$$24{,}183{,}230 + 1{,}594{,}644 = 25{,}777{,}874 \text{ lb.}$$

of steam per season.

The air supply will require steam as follows for periods of day operation, "A," Table 2, of the preceding issue:

Outside Temp., Deg. F.	Hours	Rise in Temp., Deg. F.	Lb. of Steam (118 Lb. per Hr. per Deg. Rise)
0–10	10	65	76,700
10–20	50	55	324,500
20–30	205	45	1,088,550
30–40	500	35	2,065,000
40–50	350	25	1,032,500
50–60	430	15	761,100
Total.....	1545	..	5,348,350

The total steam for heating will be

$$5{,}348{,}350 + 25{,}777{,}874 = 31{,}126{,}224 \text{ lb.}$$

Assuming 9 lb. evaporation and 2000 lb. per ton of coal, the actual coal required for heating will amount to nearly

$$\frac{31{,}126{,}224}{18{,}000} = 1730 \text{ tons}$$

Mr. Boyden states that a kitchen and hot-water service is supplied. Allowing 12 per cent. for this service, the losses in radiation and vapor and the fact that New York temperatures are used instead of Boston with a 2-deg. difference in the average, the coal would amount to

$$1730 + 208 = 1938 \text{ tons}$$

or nearly the same as Mr. Boyden's figure of 1921 tons.

It may be noticed that this system of figuring steam for heating gives very close results when the variation in operating the plant and outside weather conditions for different periods is taken into account.

By dividing the coal in Fig. 3 of Mr. Boyden's article by 5 the kilowatt-hour load is found. It shows about 60 kw. from 5 a.m. to 8 a.m. and about 150 kw. from 12 to 5 a.m.

With high pressures and compounded a 500-kw. engine ought to show a steam consumption of 22 lb. per indicated horsepower-hour. Allowing 4 lb. for losses in the engine and generator and adding one-third more for a kilowatt-hour would give 35 lb. per kilowatt-hour at the switchboard. However, when operating on the heating, 40 lb. will be allowed and 35 lb. when exhausting to the atmosphere.

Figuring an evaporation of 9 lb. per pound of coal, Mr. Boyden's chart shows a requirement of 9000 lb. of steam per hour from 2 to 5 a.m. and 12,000 lb. of steam at 6 a.m. The room temperatures were constant at about 66 deg., and did not go to 70 deg. until the fans were started, with a rising temperature out of doors. Even after the fans were started the steam consumption remained at about the same point as the direct radiation, showing excessive work on the latter during the early morning hours.

When a plant is operating on live steam and a certain amount of work has to be done, it makes small difference when it is accomplished. Where a large amount of exhaust steam is available at certain hours and for a few hours at night the supply is insufficient, it would not be necessary to turn in live steam for a short period unless the weather was below 40 to 50 deg. out of doors. In considering the isolated plant, live steam will not be used for periods above 40 deg. For night operation there will be

$$60 \times 40 = 2400 \text{ lb.}$$

of exhaust per hour for 3 hr. and

$$150 \times 40 = 6000 \text{ lb.}$$

TABLE 1. LIVE STEAM FOR NIGHT HEATING

Outside Temp., Deg. F.	Time in Days	Steam, Lb. per Hr.	Steam Required over Exhaust from 150 Kw.		Steam Required over Exhaust from 60 Kw.		Live Steam Required, per Day, Lb.	Total Live Steam, Lb.
			Lb. per Hr.	Lb. per 5 Hr.	Lb. per Hr.	Lb. per 3 Hr.		
0–10	2.7	9450	3450	17,250	7450	22,350	39,600	106,920
10–20	11	8488	2488	12,440	6480	19,440	31,880	350 680
20–30	35	7458	1460	7,300	5060	15,180	22,480	786,800
30–40	61	6430	....	.....	4030	12,090	12,090	737,490
								1,981,890

TABLE 2. LIVE STEAM FOR HOLIDAYS

Temp., Deg. F.	Hours	Steam per Hr., Lb.	Live Steam per Hr., Lb	Total Live Steam, Lb.
0–10	5	9450	7050	35,250
10–20	25	8488	6088	152,200
20–30	110	7458	5058	556,380
30–40	110	6430	4030	443,300
				1,187,130

of exhaust per hour for the 5 hr. from 12 to 5 a.m.

It will be necessary in determining the amount of live steam used for heating to find the number of days included for each period from the Table 2 previously cited when the temperature will be low enough to require it. Adding the percentage at the bottom of the columns for each period, the total time can be readily determined, also the hourly steam consumption by adding to the average the excess previously figured.

There will be ample exhaust steam week days for all purposes, but there will be requirements of live steam on Sundays and holidays, and it may be assumed that the load for power will be a minimum, or, say, 60 kw. and 2400 lb. of exhaust steam available.

Taking the hours from Table 2 of the Mar. 12 issue, "C" for holidays, and steam requirements as before, and adding 5 hr. of 0- to 10-deg. weather, the following results are obtained:

Adding the live steam for nights and holidays gives

$$1{,}981{,}890 + 1{,}187{,}130 = 3{,}169{,}020 \ lb.$$

per season. Dividing by the evaporation per ton of coal gives

$$\frac{3{,}169{,}020}{9 \times 2000} = 176 \ tons$$

per season. Subtracting from the total coal gives

$$1730 - 176 = 1554 \ tons$$

available for power. At 40 lb. steam per kilowatt-hour, 9 lb. of evaporation and 2000 lb. of coal to the ton the 1554 tons of coal will generate

$$\frac{1554 \times 9 \times 2000}{40} = 699{,}300 \ kw.\text{-}hr.$$

of electrical energy.

There are three 250-hp. boilers at present, and at 45 lb. of water per kilowatt and 1100 kw. as the maximum load, there will be required

$$\frac{1100 \times 45}{34} = 1456 \ hp.$$

in boilers. An allowance of 1600 hp. in boilers will be made, and deducting the present capacity of 750 hp. leaves 850 hp. additional to be installed. This should give a spare boiler under all ordinary conditions. A capacity of 1500 kw. in generating units should be sufficient. Allowing $30 per horsepower for the boilers, accessories and piping, and $50 per kilowatt for the generating units, with 10 per cent. interest and depreciation, will give the following expenditures in new equipment:

Engines and generators, 1500 kw. @ $50 =	$75,000
Boilers, etc., 850 hp. @ $30 =	25,500
	$100,500
Interest and depreciation, 10 per	10,050
Total	$110,550

According to Mr. Boyden's paper, 2,776,370 kw.-hr. of electrical energy were used on the premises during the year covered by the records. Allowing 40 lb. of water per kilowatt-hour would give

$$40 \times 2{,}776{,}370 = 111{,}054{,}800 \ lb.$$

of water as the annual amount used in the plant. Of this amount

$$1554 \times 18{,}000 = 27{,}972{,}000 \ lb.$$

is chargeable to heating and the balance of 83,082,800 lb. should be credited to the electrical plant. At 62½ lb. per cubic foot and $1 per 1000 cu.ft., the charge for water on the power end would amount to $1329. For labor additional to that given by Mr. Boyden, $4000 will be allowed. The amount of coal for power over heating may be found by deducting the 699,300 kw.-hr. previously determined, from the total given by Mr. Boyden. Thus

$$2{,}776{,}370 - 699{,}300 = 2{,}077{,}070 \ kw.\text{-}hr.$$

Taking 35 lb. of water per kilowatt-hour when exhausting to atmosphere, 9 lb. of evaporation and 2000 lb. of coal per ton, the number of tons of coal chargeable to the electrical plant is determined as follows:

$$\frac{2{,}077{,}070 \times 35}{9 \times 2000} = 4039 \ tons$$

At Mr. Boyden's figure of $3 per ton, this would amount to $12,117. Taking the cost of heating for the year at the amount given in the paper, $8582.29, the following summary on the yearly operating costs may be taken as approximately correct:

Heating as per Mr. Boyden	$8,582
Water	1,329
Extra labor	4,000
Coal for power over heating	12,117
Interest and depreciation	10,050
Incidentals	1,500
Total	$37,578

The cost of power and heating at present is

$$\$8582 + \$53{,}800 = \$62{,}382$$

and

$$\$62{,}382 - \$37{,}578 = \$24{,}804$$

represents the saving that would be effected per year by the expenditure of $100,000 for an isolated plant. For interest, 25 per cent. is not bad. It will pay a considerable rental on the space occupied by the plant and the owner would always be sure of a tenant.

Steam Heating Large Department Stores

I was interested in reading in POWER the reprint of the paper read before the American Society of Heating and Ventilating Engineers by David S. Boyden. It gives some interesting data on the cost of heating large department stores, and particularly the store of Jordan, Marsh & Co., in Boston.

It is apparent on reading the paper that the writer is very favorable to central-station service. While the figures and the charts are valuable, the paper is particularly interesting for what it does not state. For example: No comparison is made between the probable cost of making and the actual cost of buying electricity.

I learn from the paper that electricity was purchased at approximately 2c. per kilowatt-hour and that a total of 2,776,370 kw.-hr. was generated; that the cost of heating, including fuel, labor and all sorts of sundry supplies and repairs, was $8582.29, and this account is so carefully made that even oil and waste costing $7.35 are listed. But nothing is said about the cost of running the rest of the mechanical equipment! To what part of the operation are the services of the steamfitter, electrician, chief engineer and the elevator men charged?

The layman might add the cost of electricity to the cost of steam heat and figure that he had obtained the total cost of running the mechanical equipment of the building, and this would undoubtedly be the amount which would be compared with the actual cost of operating a private plant. Such a comparison would be unfair because the engineers of a private plant not only take care of the heating system and of the electric plant, but they take care of all these other items I have mentioned.

I do not find any charge for interest or depreciation in the cost of heating, for rental of space, for the manager's time or for the profit ratio, about which we have heard so much. I see no allowance for insurance or taxes. Surely all these are items of cost in operating the heating system, just as much as they are in the cost of making or buying electricity.

If we could get a detailed statement showing the total cost of operating the mechanical equipment of this store divided into labor and fuel and other items together with the space occupied by the boiler plant, pumps, refrigerating machinery, if any, cash blowers, etc., an interesting comparison might be made between the cost of operating this store and similar stores in New York.

I am not trying to depreciate Mr. Boyden's paper because the facts are extremely interesting, but I wish to bring out the fact that where a comparison is made between isolated-plant and central-station service, all the items entering into operating expense must be figured in both cases.

P. R. MOSES.

New York City.

In the Feb. 27 issue of POWER a mistake occurred in Fig. 2 of the reprint of my paper on "Steam Heating Large Department Stores." The May temperature shown by the dotted line should have been 62 instead of 72. The error was made in plotting the figures and will probably be obvious to most of the readers.

D. S. BOYDEN.

Boston, Mass.

Inquiries of General Interest

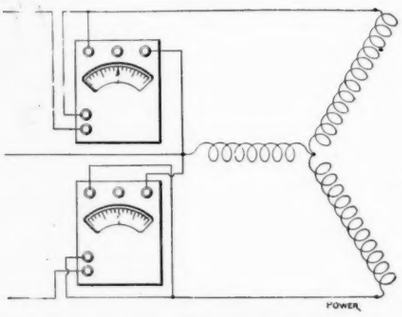
Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it

Measuring Unbalanced Three Phase Circuits

How can the power of an unbalanced three-phase circuit be measured?.

J. P. F.

In unbalanced systems two wattmeters are required. Wattmeters give the quickest and most accurate results. The method of connecting is shown in the diagram:



MEASURING THE POWER, UNBALANCED THREE-PHASE CIRCUITS

The total power transmitted is the algebraic sum of the readings of the two wattmeters. If the power factor is less than 0.50, the power is the arithmetical difference of the readings, and if it is greater than 0.50, the power is the arithmetical sum of the two readings.

Horsepower Required for a Pump

What is the horsepower required to operate a pump of 10 in. diameter, 14 in. stroke, making 80 r.p.m., with a 156-ft. total head?

P. J. N.

First find the quantity of water to be handled by the pump. A cylinder 10 in. in diameter has an area of 78.54 sq.in. and in one revolution the piston will travel

$$2 \times 14 = 28 \text{ in.}$$

The volume of water passed through the pump per minute, if the piston and valves were tight, would be

$$\frac{78.54 \times 28 \times 80}{1728} = 101.8 \text{ cu.ft. per min.}$$

It is necessary to convert the cubic feet of water into pounds in order to obtain the foot-pounds of work done by the pump per minute. One cubic foot of water weighs 62.5 lb.; therefore

$$101.8 \times 62.5 = 6362.5 \text{ lb. per min.}$$

This amount of water is raised a distance of 156 ft.; consequently, the work done by the pump per minute is

$$156 \times 6362.5 = 992{,}550 \text{ ft.-lb.}$$

or

$$\frac{992{,}550}{33{,}000} = 30.08 \text{ hp.}$$

This is the theoretical horsepower required to raise the water through 156 ft. in one minute. If the overall efficiency is 50 per cent.,

$$\frac{30.08}{0.50} = 60.16 \text{ hp.}$$

would be required.

Stage Expansion Turbine

What is a stage expansion turbine?

M. V. B.

A stage expansion turbine is one in which the steam is expanded from initial pressure down to a certain point, and is passed successively through a set of revolving vanes and a set of fixed vanes. When the velocity due to expansion is expended, the steam is again expanded through another set of nozzles to a still lower pressure and the velocity is again extracted by passing it through another set of moving and fixed blades whose length and area are a little larger than the first. This operation is continued down to the terminal pressure by successive stages. With the larger machines as high as six stages are used.

Adiabatic and Isothermal Expansion

What is meant by adiabatic and isothermal expansion?

M. C. I.

When a gas in expanding and doing work does not receive from or deliver to an outside medium any sensible heat, and, consequently, its temperature is lowered in doing work, the gas is expanding adiabatically. For the same reason, if the gas does not transfer its heat or receive any except that produced by the work of compression, it is said to be compressed adiabatically. If a gas during expansion receives heat from some other source, equal to the amount expended in doing work, the gas is expanding isothermally and if it gives up all the heat of compression, it is compressed isothermally.

Speed of an Induction Motor

What is the rule to determine the speed of an induction motor?

R. M.

The speed depends on the frequency (cycles per second) and the number of poles in the field frame, and the slip due to the part of the load actually in use. The slip or difference in ratio of a rotating field or rotor is due to the resistance opposed to the rotor current. The slip will vary from 1 per cent. in a motor designed for close regulation to 40 per cent. in one designed for a special purpose. The rule may be expressed as

$$R.p.m. = 120 \frac{\text{frequency}}{\text{no. of poles}} \times (100 - \% \text{ slip})$$

British Thermal Unit

What is the correct definition of the British thermal unit?

R. G. P.

Marks and Davis define the British thermal, or heat, unit as $\frac{1}{180}$ of the heat required to raise 1 lb. of water from 32 to 212 deg. Peabody defines it as the heat required to raise 1 lb. of water from 62 to 63 deg. Rankine's definition is the quantity of heat required to raise the temperature of 1 lb. water 1 deg. F. at the temperature of its maximum density. The Marks and Davis unit is now widely used for engineering purposes, the heat of vaporation, or latent heat at 212 deg. F., being 970.4 B.t.u. Peabody defines the heat required to raise 1 lb. of water from 32 to 212 deg. F. as 180.3 instead of 180 units, and the heat or vaporization at 212 deg. F. as 969.7 B.t.u.

Heat Units per Ampere

How much heat is developed in a wire per ampere of current flowing through it?

R. W. O.

The heat developed is proportional to the watts, not the current alone. The watts lost are equal to the resistance of the wire $\times$ the square of the current. The expenditure of one watt in overcoming resistance develops 0.000948 heat unit per second. The relation between current and heat is as follows:

$3.413 \times Amperes^2 \times Resistance = B.t.u.$

The B.t.u. thus computed are heat units per hour.

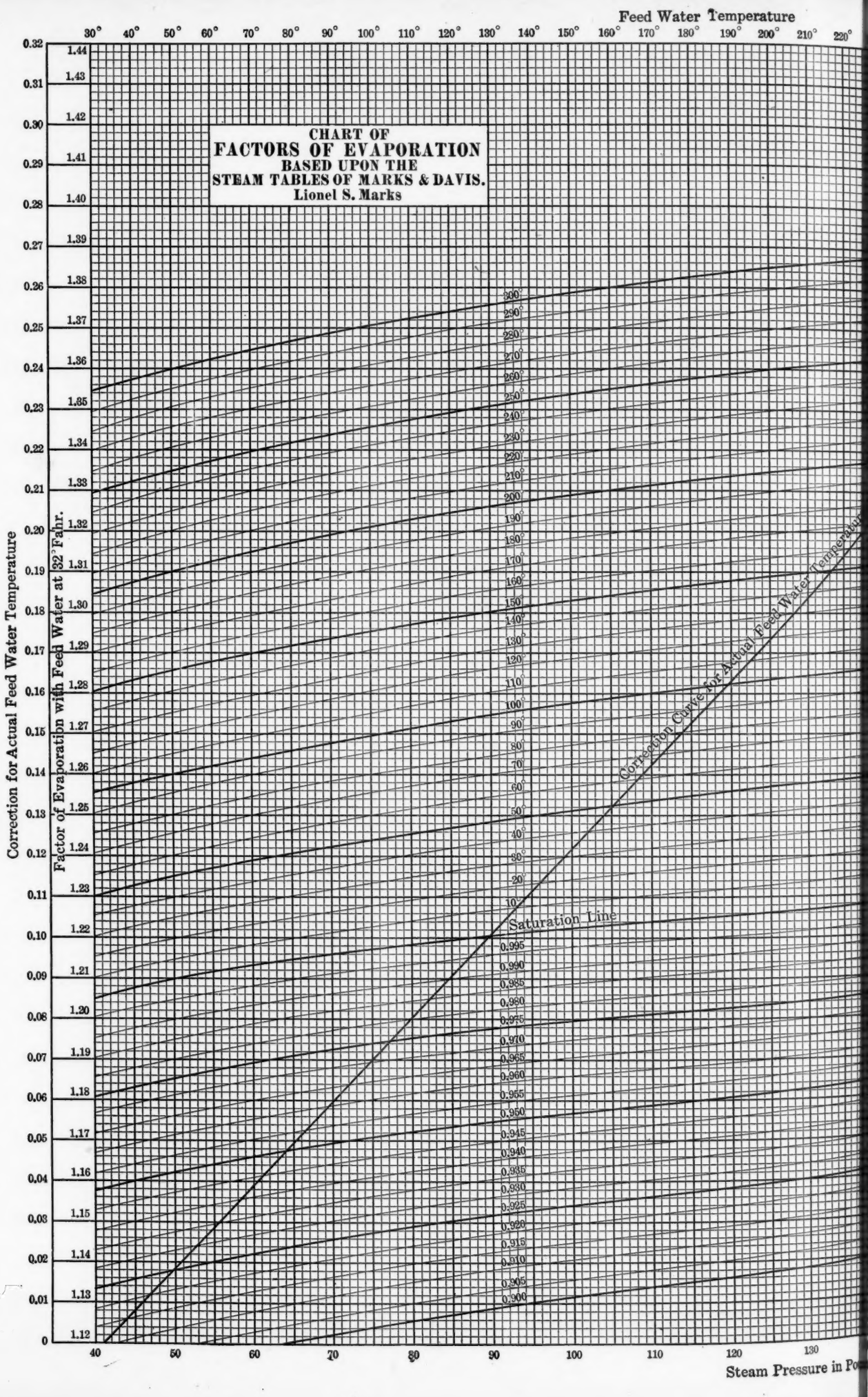
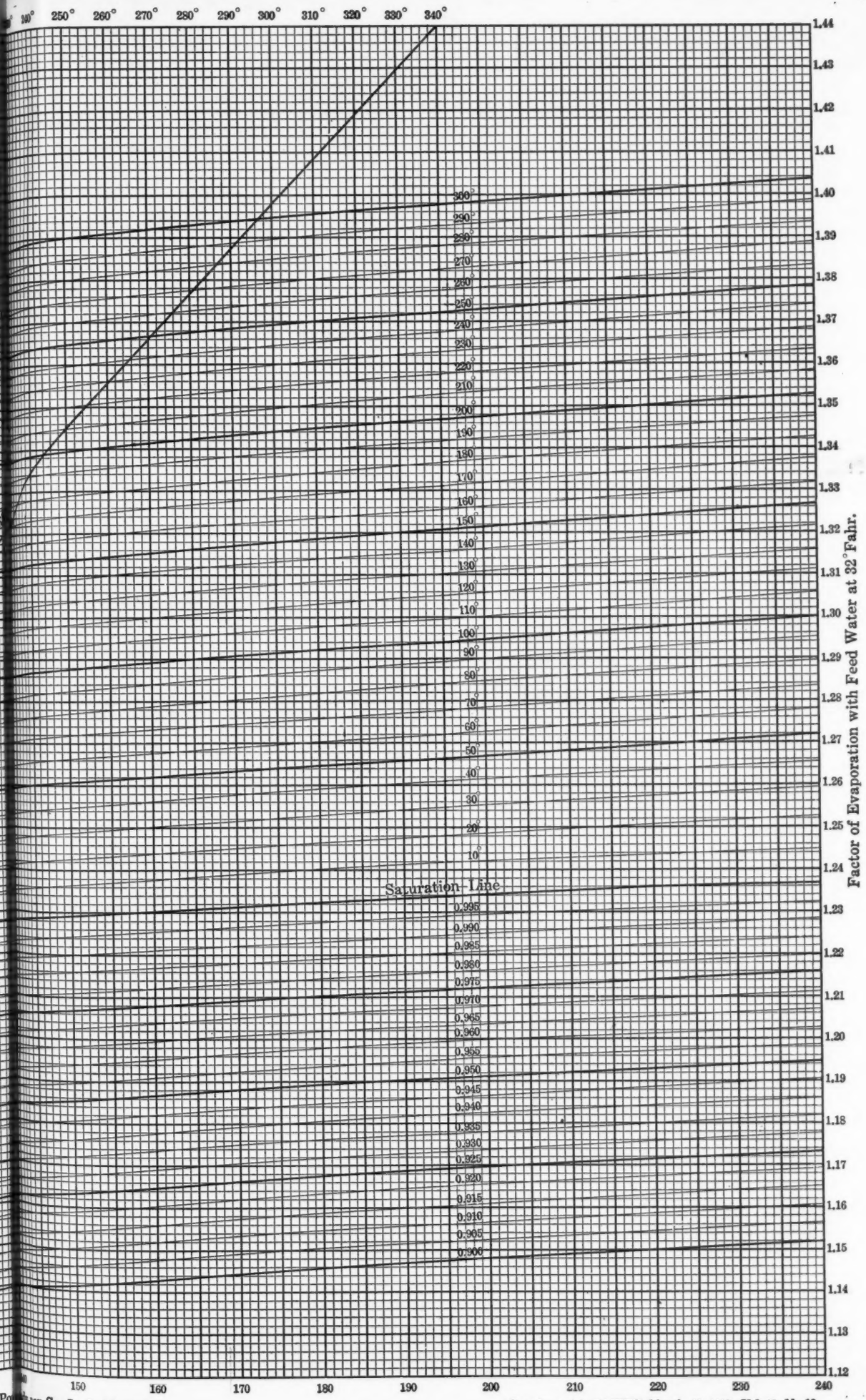
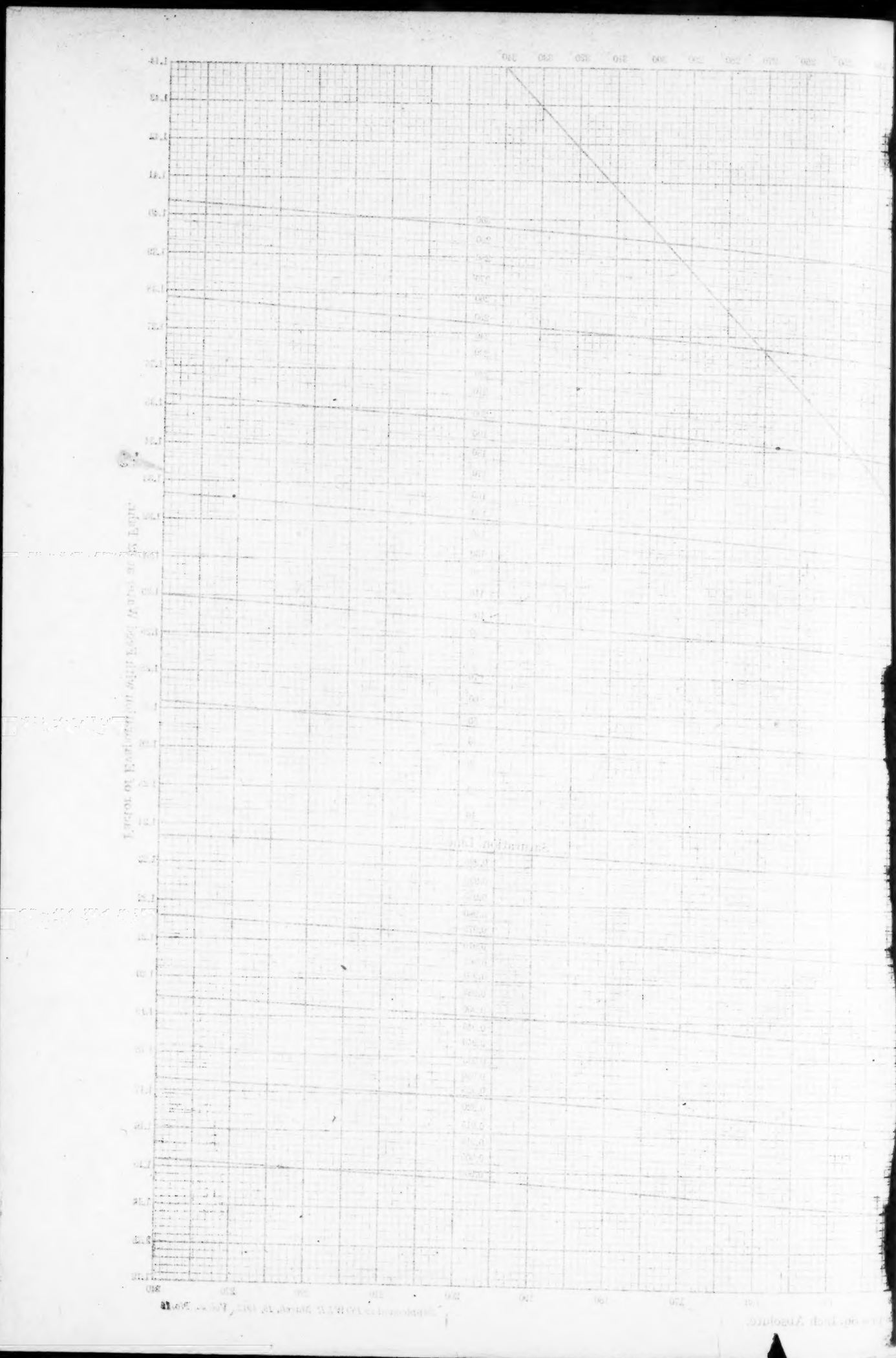


CHART OF
FACTORS OF EVAPORATION
BASED UPON THE
STEAM TABLES OF MARKS & DAVIS.
Lionel S. Marks
Feed Water Temperature
30° 40° 50° 60° 70° 80° 90° 100° 110° 120° 130° 140° 150° 160° 170° 180° 190° 200° 210° 220°
Correction for Actual Feed Water Temperature
Factor of Evaporation with Feed Water at 32° Fahr.
Correction Curves for Actual Feed Water Temperature
Saturation Line
Steam Pressure in Pounds per S


Factor of Evaporation with Feed Water at 32° Fahr.
Saturation Line
Pounds per Sq. Inch Absolute.



Issued Weekly by the
Hill Publishing Company
JOHN A. HILL, Pres. and Treas. ROB'T MCKEAN, Sec'y.
505 Pearl Street, New York.
122 South Michigan Boulevard, Chicago.
6 Bouverie Street, London, E. C.
Unter den Linden 71—Berlin, N. W. 7.

Correspondence suitable for the columns of POWER solicited and paid for. Name and address of correspondents must be given—not necessarily for publication.

Subscription price $2 per year, in advance, to any post office in the United States or the possessions of the United States and Mexico. $3 to Canada. $5 to any other foreign country.

Pay no money to solicitors or agents unless they can show letters of authorization from this office.

Subscribers in Great Britain, Europe and the British Colonies in the Eastern Hemisphere may send their subscriptions to the London Office. Price 21 Shillings.

Entered as second class matter, December 20, 1910, at the post office at New York, New York, under the Act of March 3, 1879.

Cable address, "POWPUB," N. Y.
Business Telegraph Code.

CIRCULATION STATEMENT

Of this issue 29,500 copies are printed.

None sent free regularly, no returns from news companies, no back numbers. Figures are live, net circulation.

Contents

	Page
An Interesting Power System	384
Air Compressor Efficiency	385
Chart of Factors of Evaporation	386
Notes on the Use of Belt Conveyors	387
Peculiar Sprocket-wheel Arrangement	388
What a Candidate for a Second-class License in Massachusetts Has to Know	389
5000-Pound Pressure Pump	389
Schedule of Flanges and Flanged Fittings	390
Condensers for the Power Plant	392
Low-pressure Fans	394
How Glue Stopped the Leak	395
Wainwright Water Heater	395
List of Water Equivalents	395
Increasing the Capacity of Large Gas Engines	396
Effect of Temperature on Crude Oil	400
Westinghouse Belt-driven Alternators	401
Keeping Oil Off the Commutator	402
An Improvised Three-phase System	402
Heating and Ventilating System of the Ritz-Carlton Hotel	403
An Isolated Plant Would Save $24,000	405
Steam Heating Large Department Stores	407
Inquiries of General Interest	408
Editorials	409-410
Readers with Something to Say: Condenser and Steam Jet Froze ....Oil Pipe Leaked....Freak Diagrams .... Broken Eccentric Strap .... Raising the Boiler Pressures .... Lubricated the Steam Line....Shaft Aligning Device .... Condensing versus Noncondensing .... Caring for Hot Journals .... Grease for Valve Stem Bearings	411-413

POWER

Standardizing Flanged Fittings

In the past, engineers when laying out the piping details of power plants have often been seriously inconvenienced because of a lack of uniformity in the design of flanged fittings. Also, it was believed in some instances that the flanges lacked sufficient strength to withstand the stresses imposed.

With a view to standardizing the dimensions of flanged fittings and overcoming any inherent weakness in the design, a committee was appointed by the National Association of Master Steam and Hot Water Fitters to coöperate with a similar committee from the American Society of Mechanical Engineers, for the purpose of devising a schedule which, while overcoming existing faults, would strike a mean as to the practices of the various manufacturers. The results of the joint committee's work are embodied in a schedule which will be found elsewhere in this issue.

Before proceeding to definitely fix the dimensions given in the schedule, the committee earnestly sought the cooperation of the manufacturers, not only that it might benefit by their advice and experience, but in order that the changes might cause as little trouble to them as possible. We understand, however, that some of the manufacturers neglected to respond to this invitation and the preparation of the schedule continued without their assistance. Now that it has been completed and officially adopted by one of the societies concerned, several of the manufacturers are beginning to find fault with it, pointing out radical departures from their present standards, especially in the "center-to-face" and "face-to-face" dimensions.

In view of this opposition, it seems likely that there may ultimately be some changes in the schedule. However, as any action in this respect would take considerable time, we are giving the schedule as adopted and will advise our readers of any changes that may be made in the future.

Specifications

The tendency to purchase all power-plant equipment, fuel, oil and other supplies under specification is rapidly increasing. It should not be understood, however, that this tendency is caused entirely by lack of confidence in the manufacturer of such equipment or in the seller of fuel, supplies, etc. While there has been considerable unscrupulousness displayed—and is yet, for that matter—in the effort first to get the order and second to fill it as cheaply as possible regardless of quality and workmanship, this is the exception rather than the rule. In fact, the relations between the buyer and the seller are now such that the latter will usually accompany his proposal with specifications, drawings and guarantees so that there can be absolutely no misunderstanding after the contract has been placed. The reason for this can be attributed to his lack of confidence in the buyer just as easily where the buyer writes the specification.

The great difficulty now is the fact that many, if not a majority, of the specifications are written in a rather vague, indefinite way so that there are numerous loopholes for either party to crawl out if he so desire. If a specification is worth writing at all, it should be so written as to leave no doubt as to its meaning. In the first place, it should cover the ground thoroughly and include everything that the purchaser wishes to get. For example, it is hardly fair for him to insist upon getting without extra cost some additional instrument or regulator for his switchboard that happened to be forgotten in making up the detailed specifications. Even if the seller realized the necessity for such an instrument in the first place, he could not be held liable for any more than negligence in failing to advise the prospective purchaser of the fact.

Perhaps the greatest fault to find with many of the specifications is the use of such terms as "suitable material," "ample capacity," "highest class of workman-

ship," etc. These expressions are without meaning and, consequently, have no value. For instance, cast iron may, from one viewpoint, be suitable material for a certain part, whereas the purchaser prefers steel, thus resulting at once in a controversy. It often happens in such a case that the seller has based his price on cast iron, thinking it would answer the purpose very well. As to workmanship, there is probably a standard for almost every shop in the country and the highest class of workmanship in the one filling the contract would, no doubt, technically conform to the specification no matter how poor it might be. In this respect, however, a great deal depends upon the inspector, who can very often get the desired results by the use of a little diplomacy.

Last, but by no means least, comes the question of guarantee. After the full equipment has been received and properly installed, a test, or a series of them, if necessary, should be made in order to determine whether the machinery will properly and economically do the work intended. Many plant managers and owners think this is a needless trouble and expense and are satisfied to accept and pay for the material as received, provided it does the work, without any question being raised as to low operating costs. This is false economy and is often expensive. An exhaustive test for practically every important installation made is justified as it will not only determine the exact operating cost (and thus either show a saving or otherwise on the investment), but also it will often point out where some slight alteration in the general arrangement will make for even better economy or better operation.

Waste Coal in the Refuse

Some years ago the New York school board conducted a campaign of ash burning and reported that actual economy resulted from the treatment of the furnace refuse with a solution of salt and oxalic acid and then reburning it mixed with coal. The resulting economy in fuel, though not great, was just what might have been expected, not because of any latent virtue in the salt and acid solution nor because there was any possible source of heat in the ashes themselves, but because of the large percentage of unburned coal in the refuse which was returned to the furnaces to be consumed in the second burning.

Where the furnace refuse contains large quantities of partially burned coal, as is often the case in the careless operation of steam boilers, some low-cost method of recovering the unburned carbon is greatly to be desired. Screening and hand picking are usually too expensive to warrant resorting to this means. In sections of the country where the cost of coal is high, it is said that a process of "jigging" similar to that of separating ore from rock by gravity and agitation aided by a stream of water, is successfully carried out.

However, with properly designed grates and careful firing there is no excuse for any great waste of coal among the ashes. If the incombustible in the coal were ten per cent. and ten per cent. of the refuse consisted of partially burned coal, the actual amount wasted would be not more than one per cent.

Constructive Thinking

"Do not trouble your mind about details, but rather devote your time to constructive thinking." These instructions were given to the managers of a great industrial corporation by its head.

How would this work out if the chief engineer adopted it as a policy for the running of his department? Would he be justified in leaving the details to the care of his engineers while he busied himself with constructive thinking?

Is the competency of engineers of the rank and file such as to enable the chief to repose confidence in their ability to care for the details of operation? In some plants the chief could successfully carry out this policy, but there are many where he is compelled to remain at the plant fifteen hours a day, and then perhaps be called out of bed at three in the morning to put a crippled service into commission. This condition represents a deplorable loss of human energy, a waste of time and money and very often reflects a lack of proper interest on the part of his subordinates or the management.

Accidents are bound to occur, but their number can be greatly reduced by constant care and vigilance exercised by engineers and managers alike.

There are many industrial concerns which have introduced efficiency systems into the manufacturing of their products, but they have neglected to recognize the value of efficiency in the operation of their power plants. That the power plant is not classed as a productive division is no good reason why it should receive less consideration than those departments directly engaged in the production of the saleable goods. As soon as managers become aware of the possibilities for increased efficiency in this direction and find out that a saving or loss their is a saving or loss in the machinery of production, the power department will have come into its own.

It is the duty of a chief engineer to point out how this may be accomplished, and he will be able to do so if his subordinates are competent to take care of the details and give him the opportunity to think out the problems of operation—to think constructively. If the engineer will learn to read intelligently the charts from recording watt, volt, steam-flow and temperature meters and understand other means of obtaining available data, he can quickly locate the inefficient spots in the plant, and his department will without much doubt receive the attention it deserves at the hands of the management. To do this in a large plant, however, the chief must be relieved of the petty operating details, he must become a general and cease to be merely a maintenance man.

Education plus "overalls" equals a good operating engineer. This is a simple success formula, but many a college-bred man has been parachuted off the top of the theory ladder into the grime of the boiler room before he could get the correct answer.

The unsettled condition of the coal industry should furnish an incentive for the advocates of peat to prove its worth as a substitute for coal in those places where it is commercially available.

Regard no man as so inferior but that he can contribute to your fund of knowledge, says *Coal Age*. If nothing else, one can learn a lesson from his failure.

Vermont is the only state which had no boiler explosion in 1911.

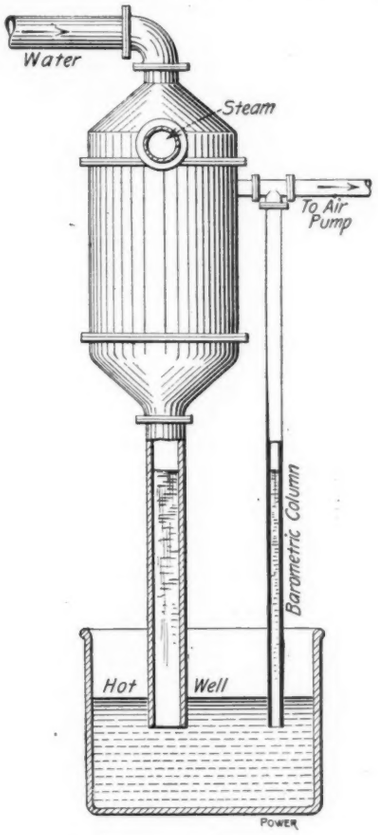
Readers with Something to Say

A letter good enough to print will be paid for. Ideas, not mere words, wanted

Condenser and Steam Jet Froze

Following are two interesting instances of the troubles encountered in our plant during the recent excessively cold weather. Some may be inclined to be skeptical; nevertheless, they actually occurred.

I have a jet condenser, constructed something as shown in the accompanying



CONDENSER PIPING

illustration, that serves a 3500-hp. engine. There is a barometric column connecting into the line running to the air pump at a point just outside the condenser.

One day the condenser lost its vacuum for no apparent reason. Investigation showed that the water in the barometric column had frozen and that steam had condensed and frozen in this pipe, building up a column of ice that finally filled the pipe leading to the air pump, causing the vacuum to fail.

About the same time I had in use in one of the boiler houses steam blowers supplying forced draft to the boilers. On one of the cold days it was noticed that one of the boilers was not steaming well. The steam from the blowers had condensed and had built up a mass of icicles in the space surrounding the jets, practically cutting off the air supply.

To one not familiar with conditions here, where the temperature remained between zero and 15 deg. below for two weeks, these troubles may seem incredible.

J. F. MOWAT.

Joliet, Ill.

Oil Pipe Leaked

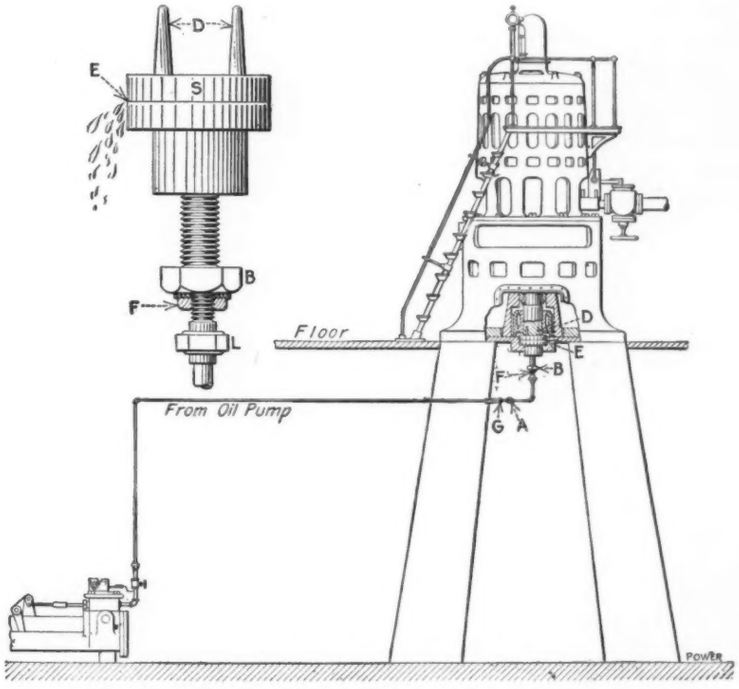
The oil feed pipe which screws into the bottom shoe of a step bearing in a vertical shaft steam turbine is often a source of trouble.

Several days ago the oil pressure dropped from 225 down to about 200 lb. per square inch. This was caused by the lowering of the steam pressure on the line and consequently the pump operated more slowly, which in turn lowered the oil pressure under the step, and both shoes naturally got a little rubbing.

The plant is not equipped to prevent these occurrences, so the men have to be on the watch. In this case it was necessary to remove the step and replace it with a new one from the shop.

After starting up, the pressure was maintained satisfactorily, except that a small leak developed where the feed pipe screws into the bottom shoe. The leak increased and upon opening up the union and removing the pipe, I found that it had been holding by two threads. I replaced and screwed it in as much as I could, giving it four complete turns, and thought that this would stop the leak, but the next morning it was leaking again.

For a remedy I devised a scheme that will be easily understood by referring to the illustration. At *F* is shown the leak; *A* is a union and *G* a baffle. I removed the short pipe *F*, threaded it and shortened it as shown in the enlarged drawing. Then I slipped a leather washer over the threaded portion of *F* and then a brass nut, which was screwed up tight against the bottom of the adjusting screw *B*. This made an excellent packing and stopped the leak.



WHERE THE STEP-BEARING OIL PIPE GAVE TROUBLE

To make things clearer, *D* shows the two pins which fit in the bottom of the turbine shaft. The revolving step plate *S* therefore is removable and in case of wear can be replaced. *E* is where the liquid comes out between the top plate and

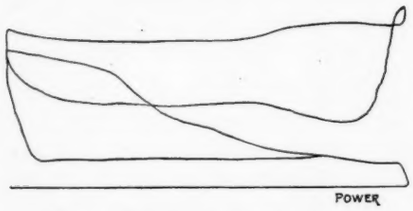
the bottom supporting plate. I have changed the union from *A* to *L*.

LUKE MARIER.

Fall River, Mass.

Freak Diagrams

The accompanying indicator diagrams taken from our Corliss engine may be



WHAT CAUSED THESE DIAGRAMS

interesting as freaks. Who knows what was the trouble?

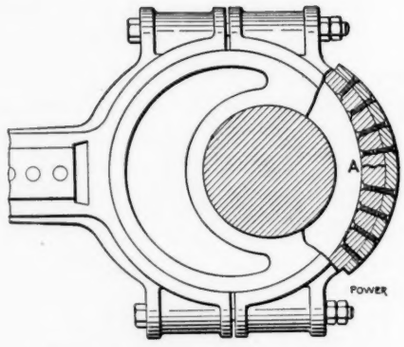
W. CLARK.

Penn Yan, N. Y.

Broken Eccentric Strap

While operating a locomotive crane, one of the eccentric straps broke in a peculiar manner at about the point represented by *A* on the drawing. This, of course, crippled the engine and needed immediate attention.

I forged out a steel band strap to fit on the outside of the broken parts; then I drilled four holes in the broken strap on either side of the break and drilled corresponding holes on the forged piece. The holes were countersunk on the in-



ECCENTRIC STRAP AS REPAIRED

side of the old strap and also on the outside of the forged band. The holes in the band were tapped with a $\frac{3}{16}$-in. tap and flat-headed stove bolts screwed tightly in place. The ends protruding on the inner side of the strap were riveted over as much as possible to fill the countersunk holes. A scraper was then employed to smooth off any protruding edges.

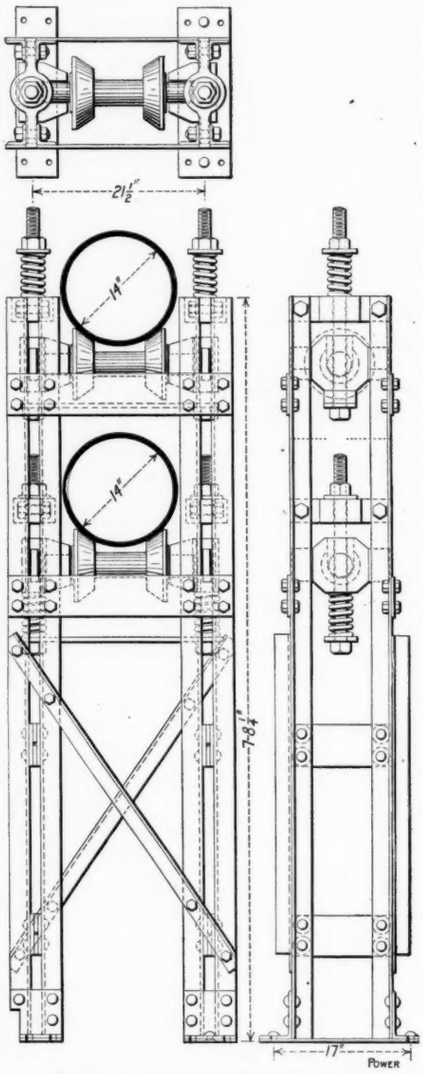
The strap was replaced and gave good service until the machine was abandoned for one of greater capacity.

F. F. SENGSTOCK.

Chicago, Ill.

Pipe Support

The accompanying sketch illustrates a support for two 14-in. high-pressure steam pipes. The pipe is situated so



DETAILS OF THE PIPE SUPPORT

that it expands downward horizontally and sidewise. The support takes care of the two 14-in. pipes, and is located in the center of the pipes which are 30 ft.long.

When finding the carrying capacity, the weight of the pipe plus the water in it was considered. The support serves its purpose in every detail, and avoids the leaky joints occasioned by the solid support formerly used.

R. G. Cox.

Milwaukee, Wis.

Raising the Boiler Pressures

In many cases it has been found that damage to property and loss of life have been caused by increasing the pressures of steam boilers.

This should be carefully considered before it is done, know that the boiler heads are braced sufficiently to withstand the increased pressure and that the longitudinal seams are of proper design. The strength of the entire boiler should be figured and allowance made for age and weak spots. The steam piping through the plant must also be examined.

I recently entered a plant which contained three 72-in. by 19½-ft. boilers and the plant looked safe and well laid out, but I soon noticed that a plate was fitted over the bonnet of the feed valve and that U-shaped bolts had been put around the body of the valve to clamp the plate down over the bonnet to hold it tight to the main body; other valves were also harnessed in the same way. I was told that a bonnet had let go some time before, as it was too light for the 120 lb. pressure which was carried.

When the plant was originally constructed a few years ago it was intended to carry but 85 lb. steam pressure, although the boilers were to carry 140 lb. pressure per square inch. The blowoff pipe was fitted with extra-heavy stop-cocks with a low-pressure gate valve between the stop-cock and the boiler. Both valves were neatly harnessed to prevent the bonnet from blowing out.

The pressure had been raised a few pounds at a time by each engineer as the load increased, from 70 lb. until the boilers were working under a 120-lb. head. On discovering the low-pressure fittings, the pressure was again lowered to 110 lb. per square inch. I think if more caution was taken in such cases, fewer accidents would occur.

R. A. CULTRA.

Cambridge, Mass.

Lubricated the Steam Line

In the plant where I am employed trouble occurred from many leaks in the steam mains. As there was no time to shut down for repairs, I decided on a plan to stop them. There were about 40 leaks between the boiler house and the engine rooms, but today not a leak is to be seen.

No new gaskets were used, but the leaks were all stopped by feeding oil into the mains with two lubricators, one of which was made of an old trap, the other out of a 6-in. pipe, 4 ft. long. The oil was fed in as fast as the lubricator would work at the start, about 5 gal. every 24 hr. The more oil fed the quicker it will reach the joints and stop the leak, as it takes some time before the oil reaches all of the joints in a large plant, unless several lubricators are used, because the pipe must get coated first. At the present time we use 1 gal. of oil every 24 hr. to keep the lines well coated and prevent further leaks. It took about six months to stop the leaks.

When using oil it will first be noticed that the small leaks will disappear as the

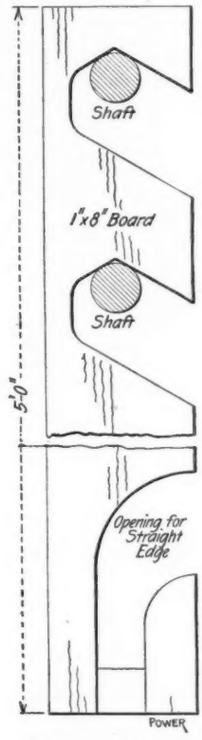
oil reaches the joints, and so on until the leaks have been stopped.

CHARLES H. GREENE.

Bath, Penn.

Shaft Aligning Device

The accompanying is a sketch of a home-made apparatus for lining up shafting. Nail two 1x8-in. boards together and saw them, as shown; then make a straight-edge about 8 ft. long. With this device one can align any shaft from ½



ALIGNING BOARD

to 5 in. in size. The two holes are so that if anything should conflict with the straight-edge it can be adjusted from one hole to the other.

ALLEN J. STOCKS.

Seattle, Wash.

Condensing versus Non-condensing

In a plant of approximately 30,000 hp., the chief engineer and the assistant say that five of the cross-compound engines, which are of 3000 hp. each, and are supposed to run condensing, will pull more load running noncondensing than they will when operating condensing. The chief also says that the funny thing of it is he cannot keep up steam when running noncondensing.

I have seen one of these engines deliver 5060 hp. to the switchboard before the condenser fell down, but how much would that engine pull now without the condenser? I cannot understand in what shape the valves are set to get the results the chief mentions, as when the vacuum is removed from the 64-in. low-pressure piston it is necessary to run the receiver pressure up to at least 30 lb., which puts an excessive back pressure against the high-pressure piston, cutting down the efficiency of that cylinder.

A man might just as well say that he can pull more load with a simple engine with 140 lb. of steam than he can with 150 lb. I do not know how the engineers feel about the balance of the plant. I wonder if they would do this same thing with a turbine.

C. L. JOHNSON.

Newark, N. J.

Caring for Hot Journals

Many and varied are the ideas for cooling a hot journal, but the main thing is to cool the journal without stopping the plant if possible.

The method I usually employ for a very hot journal is to first pour in some black cylinder oil, and if graphite is handy mix some with the oil. The heavy oil will lubricate—while an ordinary oil would simply boil—and if the shaft is rough the graphite will help to smooth it up.

I next slowly pour some clean, cold water through the oil hole onto the journal; I do not try to cool the journal all at once, I use the water and the cylinder oil and graphite alternately. This will usually produce a clean, cool journal.

If practicable, the cover and pedestal nuts may be eased to equalize the pull, if the journal is overloaded. By running the water through the journal it goes at once to where it is most needed and cools the inside and leaves the outside hot. The heavy oil forms a film on the shaft, and the water absorbs the heat.

I have tried sulphur, grease, white lead and castor oil, and if the journal is not very hot, these may keep it going and allow it to gradually cool down; but where there is a risk of shutting down, I prefer the cylinder oil, and it is usually at hand in a steam plant.

Using a heavier oil will help in a case like this. If the journal tends to heat, a good idea is to have an endless band hung over the shaft if possible, the lower end of the band running in a pail or tray of water. The band acts as an endless conveyor to lift the water to the shaft; this arrangement also keeps the place clean and gives no trouble.

It is advisable to examine an overheated journal at the first convenient stop and remove any brass dust or rough places on the shaft. If the shaft or bush is scored the journal should be draw-filed and polished; the brasses should also be touched up.

W. BURNS.

Govan, Scotland.

Grease for Valve Stem Bearings

One make of engine I operated had excessive travel of the various parts of the valve-gear, which necessarily caused more pin wear than would have been the case had the parts been properly laid out. The valve-gear wore more in the course of a year of steady running than a properly constructed gear should have in a ten years' run, the attention to lubrication and running being equal.

The engine was poorly equipped for lubrication. Even when carefully adjusted the governor spindle would wear down in the step bearing and it had to be set up once or twice in a summer. Oil would come to this bearing only from oil holes in the weight of the Porter style governor, through which oil was guided to the spindle.

Undoubtedly a great improvement was effected by running the governor gear in an inclosed box. The governor-pulley shaft was to run 127 r.p.m. and was only equipped with two small oil holes near the end of the bearing. These were replaced by a self-feeding cup in the center of the bearing.

The valve stems are a much neglected part of an engine. Even if an oil hole is supplied to each bearing, the hub of the bell crank sliding over the valve-stem bearing and oil hole in it is liable to cut off the supply of lubrication and take more than its share. As they are hot, on account of the proximity of the steam-valve port, I placed a grease cup in the hub of the bell crank, and also drilled four holes in the center of the valve-stem bearing and filed them out into one long slot. Now, no matter in what position the grease cup is, the lubricant will always drop down onto the valve stem.

I also cut an oil groove in the upper part of the bearing, on the inside, thus allowing the lubricant to spread over its length. Then I placed a grease cup on the wristplate and one on each exhaust-valve bearing, after taking out the small oil cups supplied with the machine, which were not large enough to give proper lubrication.

Due to these changes the engine has been kept in better shape with less attention than before the changes were made.

G. BERNA.

St. Louis, Mo.

The New York Public Library has recently organized a reference department containing catalogs of various manufacturers. This should be of great service to engineers as well as architects and contractors, and it is to be hoped that they, together with the manufacturers themselves, will coöperate in making this new department a success.

Diesel Engine at the South Pole

The "Fram," the ship in which Capt. Roald Amundsen sailed on his latest voyage of discovery to the South Pole, is propelled by a Diesel engine.

Capt. Amundsen selected this type of engine because the capacity of the "Fram" was too limited to carry a heavy burden of coal. With crude oil, the fuel of the Diesel engine, Capt. Amundsen was able to make a voyage longer by three to four times than with an equal weight of coal.

Four Men Killed in Boiler Explosion

On Mar. 7, at P. H. Gilpin's woodhandle factory at Greentown, Penn., a boiler explosion wrecked the plant, valued at $25,000, and instantly killed four men and seriously injured another.

Beyond the fact that the fatality occurred two minutes after the fifty employees had begun work in the morning, no further details are known and the cause has not yet been determined. An early account of the explosion will appear in POWER.

The American Society of Naval Engineers

The date of the annual banquet of the American Society of Naval Engineers was set this year upon March 9 that the occasion might be made commemorative of the battle between the "Monitor" and the "Merrimac." Walter M. McFarland, of the Babcock & Wilcox Co., speaking on "Naval Engineering from the 'Monitor' to the 'Oklahoma,'" told of the significance of that battle and its effect in making engineering paramount in the navies of the world. Hon. W. C. Redfield responded to the toast, "A Powerful Navy at once the Guarantee and Exponent of a Nation's Prosperity," Hon. Albert Estopinal to "The Navy and Congress," Hon. George von L. Meyer, secretary of the Navy, to "The Importance of Naval Engineering" and Hon. Joseph Cannon to "The Navy and the People." George H. Westinghouse and Col. E. D. Meier, two past presidents of the American Society of Mechanical Engineers, occupied places at the speakers' table. Capt. William Strother Smith acted as toastmaster.

Electric Propulsion to Be Tried by Navy

A system of electric propulsion will be given trial on the United States naval collier "Jupiter," which is under construction at the Mare Island navy yard in California.

In this system it is proposed to install one turbo-gtnerator running at a maximum speed of about 2000 r.p.m. and delivering current at 2300 volts to induction motors, one on each propeller shaft. The power from the turbine will thus be transmitted to the propeller shafts, electrically, with a speed reduction ratio of 18 to 1. The efficiency of transmission is expected to be about 91 per cent. The water rate for this installation is stated to range from 12.15 lb. per shaft horsepower at 14 knots, to 15.55 lb. at 10 knots.—*Engineering News.*

Considerations Governing the Selection of a Modern Steam Engine

The Harrisburg Foundry & Machine Works have issued, presumably for gratuitous distribution, a pamphlet bearing the above title. The water rate for four types of engine at 125 lb. initial pressure and atmospheric exhaust is given in the accompanying table.

Assuming that the engine runs 3000 hr. per year, 600 hr. at each of the given ratios of its capacity, the bulletin deduces that either of the compounds would be out of the question on account of their high water rate at the lighter loads; that with $3.50 coal the simple Corliss should be selected, but that with coal at $1.50 the saving might not warrant the increased investment over that required for a single-valve engine. A larger load factor would, of course, modify this conclusion.

STEAM PER INDICATED HORSE-POWER-HOUR, POUNDS

	Load				
Type of Engine	¼	½	¾	1	1¼
Simple single valve	35.3	30.2	28.5	28.1	28.4
Simple Corliss valve	32.4	24.5	23.1	23.6	24.7
Compound single valve	42.4	28.2	24.3	23.5	24.0
Compound Corliss valve	41.2	28.0	22.5	21.1	21.3

Annual Exhibit at Pratt Institute

The annual exhibit of the work of the evening classes in Pratt Institute will be held on Friday evening, Mar. 22, from 8 to 10 o'clock, and is open to the public.

The institute offers courses for men and women in drawing and design, applied art, applied science, sewing, dressmaking, millinery, household arts and various mechanical trades, and this exhibit affords an opportunity of viewing the work of the students, and of inspecting the results and methods as well as the school's equipment and general facilities for this kind of industrial training.

The work of the School of Science and Technology should prove of especial interest to men engaged in technical and trade pursuits, as it provides instruction in industrial electricity, technical chemistry, mechanical drawing and machine design, strength of materials, stationary engineering and power-plant machinery, machine work and tool making, carpentry and building, pattern making, sheet-metal work, plumbing, etc.

NEW PUBLICATIONS

CONSTRUCTION AND WORKING OF INTERNAL COMBUSTION ENGINES. By R. E. Mathot. Translated from the French by W. A. Tookey. Published by D. Van Nostrand Co., New York. Cloth; 550 pages, 6x9 in.; numerous illustrations and tables. Price, $6.

According to the subtitle, this is a practical treatise upon methods of construction, with calculations for the use of engineers, manufacturers and users, and a critical study of present-day types. The author further explains in the preface that his object in writing the book was not merely to present an exposition of the state of the art with respect to constructional details, this having been adequately accomplished in other recent publications; his purpose was to lay down "the precepts of theory and of practice as far as experience permits at the present day." Again, "he (the author) has reviewed the well known arrangements adopted by the chief gas-engine builders that in a sense have become standardized but, as far as possible, has avoided giving descriptions that have already been made public by several authors."

The average observant reader would undoubtedly have formed a much better opinion of the book if the foregoing prefatory announcements had been omitted, because the contents of the book are not in harmony with them. For example, in chapters IV, VI and VIII appear pictures of the Foos, Olds, Riverside, Warren, Stockport, National, Premier, Tangye, Ehrhart & Sehmer, Deutz, Körting, Nürnberg, Oechelhauser, Sargent and Diesel engines which are old, familiar friends, having appeared in many textbooks and most of the engineering periodicals years ago. The uptodateness of the book is rendered open to question by the inclusion of a two-stroke-cycle Buckeye engine which was built experimentally and abandoned three years ago, the Sargent engine, which has not been built for two years, the Rathbun engine with a valve-gear that was superseded about 2½ years ago, and several governor gears and ignition systems that were widely familiar ten years ago and became obsolete about four years ago. Moreover, the reviewer is unable to find anywhere in the book the "critical study" promised in the subtitle.

However, chapter XII, on details of construction, is an excellent compilation of practical formulas and other data; chapter XIII, relating to moving parts, contains some very useful material, though mixed with a mass of partly

superfluous and partly perfunctory statements (for example: "No lateral movement of the flywheel [on the shaft] can be permitted; therefore, extreme care is required in keying to the shaft." Again: "The pistons for double-acting engines consist of (1) piston; (2) rod; (3) crosshead; (4) water-cooling system"). Chapter XIV, devoted to engine testing, also contains some highly useful suggestions, but it is laden with much more elementary discussion than is appropriate in a book of this class. The next chapter on indicator diagrams comes nearer being original than any part of the book, but the final chapter, on dimensions and classifications of engines and test records is the most valuable, because it presents some results of actual experience.

The book as a whole is characterized by a decided tinge of perfunctoriness—many words conveying little tangible information; where definite, clean-cut statements of opinion or observation are made, they are often so dogmatic as to be inapplicable to general cases.

THE TESTING OF MOTIVE POWER ENGINES. By R. Royds. Published by Longmans, Green & Co., London, 1911. Size, 5½x8¾ in.; 396 pages; 193 illustrations and several tables; cloth. Price, $3.

This book is intended for engineering students who have already acquired an elementary knowledge of motive-power engineering and who desire information on the practical testing of motive-power engines. The author hopes that the work will also be of service to the engineer in practice.

As it is well nigh impossible to touch upon every phase of motive-power engine testing in a single volume of this size, little or no attempt has been made to describe the mechanical features of the different types of engine or to interpret the action of an engine from a consideration of indicator diagrams as these portions of the subject are dealt with more or less fully in several textbooks. Numerous references are given for the benefit of those requiring a more detailed account of particular subjects.

The initial chapter deals with general principles, definitions, units, etc., and is introduced mainly for the purpose of reference. The subsequent chapters take up the measurement of pressure, temperature, etc.; the measurement of brake horsepower, in which several types of power brakes are illustrated and discussed; the testing of locomotives, motor cars, etc.; the testing of steam engines and turbines, boilers, condensers and air pumps, internal-combustion engines, gas producers; refrigeration tests; the testing of air compressors, air motors, etc., fans and blowers, water turbines and pumps.

In all cases, the basis of efficiency is defined and the formulas for ascertaining them are given. Particular attention is drawn to the variable conditions under which a plant may operate and the necessity for systematic arrangements where a series of tests are contemplated.

The book gives evidence of the author's thorough understanding of the subject and it may safely be recommended as complete and accurate.

MECHANICAL WORLD ELECTRICAL POCKET BOOK FOR 1912. Published by Emmott & Co., Manchester, Eng. Boards; 200 text pages, 4x6 in.; 80 illustrations; many tables. Price, 6d.

This year's edition of this well known little book contains some 16 pages more than the previous edition. These are divided up between the rewritten sections on motor starters, transformers, lamps, electric lighting, and aluminum conductors; a new section on testing high-tension apparatus, and some new tables on power factors, aluminum cables and carrying capacity of copper conductors. The sections on dynamos and motors, alternating-current systems, alternating-current machinery and electrical measuring instruments remain inadequate to the point of uselessness. The tables constitute the most useful portion of the book.

In a previous review it was erroneously stated that the book contained new sections on verniers and micrometers, roller bearings and other nonelectrical subjects; this error was due to having glanced through the mechanical pocket-book also, prior to reviewing this one, and thereby confusing the features of the two.

POWER PLANT TESTING. By James Ambrose Moyer. Published by McGraw-Hill Book Co., New York, 1911. Size, 6x9 in.; 422 pages; 271 illustrations and numerous tables; cloth. Price, $4 net.

This book seems to be little more than a compilation and a poor one at that. The subjects treated are of the usual list found in books on experimental engineering, and the reviewer has searched the volume in vain for one chapter that smacked of originality. Compilations have a very legitimate and useful function and, when properly made, they are of real value.

The trouble in the present case is that in some chapters useless material has been put in for apparently no other reason than that of filling up space, while in other chapters important material has been omitted. To illustrate: In chapter five there are two figures of and some text matter on Watt's original steam-engine indicator which might easily be dispensed with in favor of more elaboration in other directions. In chapter nine, on flue-gas analysis, space is given up to the description and illustration of an automatic instrument which has found but little, if any, application, in this country at least, while the author neglects completely to say a word about the Gas Composimeter, which is a widely used and most successful instrument.

Many of the illustrations are old friends; we have seen them before, in textbook, government bulletin and technical paper.

PERSONAL

George H. Smith & Co., of Baltimore, Md., have opened an office in the Munsey Building as consulting and contracting electrical engineers.

J. A. Struthers, formerly electrical engineer with the Westinghouse Electric & Manufacturing Co., Pittsburg, Penn., has accepted a position as designer for the Henry Manufacturing Co., New York City.

Barton R. Shoover, formerly electrical engineer for the Youngstown, Ohio, district of the Carnegie Steel Co., of Pittsburg, Penn., has been appointed superintendent of the new Brier Hill Steel Co., Youngstown, Ohio.

Charles E. Downton, representing the Westinghouse Electric & Manufacturing Co., recently lectured before the senior class of the College of Engineering of the University of Illinois, setting forth the advantages offered by his company in the matter of positions for technical graduates.

Prof. J. G. D. Mack, the exchange lecturer from the University of Wisconsin, gave two lectures before the students and faculty of the College of Engineering of the University of Illinois on Feb. 20. One lecture was on the engineering features of the Wisconsin Tax and Public Utilities Commissions. Prof. Mack is a member of the engineering staff of the commissions. The second lecture was on some features of ancient engineering work and their relations to present-day practice, and was, in part, an appeal to engineering students to study the history of engineering as well as current practice.

C. C. Owens has recently assumed charge of the Detroit district sales office of the Westinghouse Electric & Manufacturing Co., with the title of district manager. Mr. Owens was born in 1877 near Annapolis, Md., and was graduated from Columbian University, Washington, D. C., with the degree of bachelor of science in electrical engineering. He entered the employ of the company in 1896, taking the apprenticeship course for three years, after which he spent four years in the engineering department, specializing on switchboards and controllers. For the last eight years he has been connected with the New York sales office, having had charge of the industrial and power division for the two years previous to his transfer to Detroit.

Moments with the Ad. Editor

This week we are printing two more letters in answer to the questions asked in the issue of January 23.

The first is from Massachusetts and the second from far-off Italy.

POWER never makes much of a "holler" about its foreign circulation but it's there, as friend Peterson of the Richardson-Phenix Company would say, "with bells on and its hair done up in a braid."

We are especially glad that three of our subscribers in Europe answered the editorial of January 23.

It argues well for the interest in POWER advertising abroad.

H. W. Appleton says:

The J. M. Sea Rings Packing, to my mind, is the ad. most worthy of investigation. The rings seem to be built on a plan entirely original, and also appear the best thing yet for the purpose as the construction admits of the least possible gland pressure and friction on the rods. They are put out by a concern known to be reliable, and when my present supply of piston rod packing runs out I shall requisition a supply of Sea rings.

In the small industrial plant, fire-brick is a commodity purchased only at rare intervals, and usually the concern handling them in the locality where required gets the order regardless of quality. The Harbison-Walker people, however, have a way of showing views of the different processes in the manufacture of their brick which prove to be interesting as well as making a distinctive ad.

Consequently, when fire-brick are mentioned, I can at once see a picture of some of the different parts of their works. Therefore, their ad. may be said to have the best "storing-up" value.

The Nelson Valve ad. is the best, because it attracts the attention, is clear, descriptive, without being tedious, contains no knocks on other makes, and gives the impression of a high-grade product.

I have answered the Keystone Grease ad.

It is evident that the J. M. Sea Ring ad. made the impression that was hoped for by the writer of it. It is indeed a well designed and well written ad.—a regular Low and behold ad.! (F. J. please answer.)

Mr. Appleton's description of the Nelson ad. is good stuff—right to the point without a wasted word.

E. S. Wylde is mill manager for a well known Italian concern. He gets right down to business this way:

The Taylor Stoker ad. prompts immediate investigation because, although I am not the engineer of this concern, I am the mill manager, and this ad. has made me see in a way I have never realized before the enormous coal waste that must take place with black smoke being allowed to issue from the chimney, and therefore makes me wish to investigate it immediately and put an end to an ugly sight and save the money that is evidently being wasted.

The ad. of the Vance Steam Trap has the most educative value because the six advantages set forth in the ad. bring home to one the fact that a number of parts go to make the complete trap, and this helps to make one realize the need of educating oneself to understand and appreciate the countless parts that go to make the complete power plant.

To my mind the three great points in an ad. are: First, that it should catch your eye; second, that it should hold your attention; third, that after having read it, it should so recommend the thing advertised that one wishes to possess it.

Therefore, I consider the ad. of the International Textbook Company the best, catching one's eye as it does by its size and contrast with the next page, holding one's attention as one reads it, setting forth the advantages of "The Greatest Mechanical Library in the World," and thirdly, making one wish to possess it, covering as it does the whole field of mechanical engineering from raw materials to finished articles and their application.

I answered the ads. of Albany Grease, the Taylor Stoker and the International Textbook.

If all ads. could be and were written to Mr. Appleton's prescription there would be less talk of advertising wastes and more of advertising profits.

Let us get the stuff into space that attracts, holds and convinces. Then we can all do business on the Albany Grease principle—with minimum friction.

NEW EQUIPMENT

ATLANTIC COAST STATES

Chicopee, Mass., will expend about $96,000 on improvements to the municipal electric-light plant. Four turbines and three boilers will be installed.

The county commissioners are considering the installation of an electric-lighting plant at the county jail at Worcester, Mass.

Woodbury, N. J., is considering the installation of a municipal electric-light plant.

E. F. Williams, superintendent of water department, Duquesne, Penn., has recommended the installation of a municipal electric-light plant.

Martinsburg, Penn., is considering the installation of a municipal electric-light plant.

The city council, New Castle, Penn., is considering the installation of a municipal electric-light plant. Estimated cost, $133,000.

The Beehive Building, Greemount and Preston Sts., Baltimore, Md., will install generators to produce 2000 hp. Estimated cost, $40,000.

Centerville, Md., is considering the construction of a municipal electric-light plant.

SOUTHERN STATES

Whitakers, N. C., is considering the installation of a municipal electric-light plant.

N. E. Whitman and associates are considering the installation of an electric-light plant at Lewisville, N. C.

Cheraw, S. C., is considering an expenditure of $8000 for improvements to the municipal electric-light plant.

Walterboro, S. C., contemplates the installation of an electric-light plant to furnish electricity for a street-lighting system.

Waterloo, S. C., is considering the installation of a municipal street-lighting system.

Boaz, Ala., will install a municipal electric-light plant. W. H. Barlett is mayor.

Covington, Ky., is contemplating the installation of a municipal electric-light plant.

CENTRAL STATES

We are advised that Shuler & Benninghofen, Hamilton, Ohio, will not build a power plant in connection with their woolen mills, as noted in the issue of Mar. 5, but will install an Otto gas engine in the main building of their plant.

The Wise Electric Co. will construct a power plant at Mowrystown, Ohio. The proposed plant will be equipped with a 75-kw. generator, a 125-hp. engine and a 16x72-in. boiler. Estimated cost, $10,000.

Rogers, Ohio, is contemplating the installation of a municipal electric-light plant.

Bids will be received by W. J. Springborn, director of public works, City Hall, Cleveland, Ohio, until Mar. 14, for transformers for the municipal electric-light plant.

The Oakwood Street Ry. Co., Dayton, Ohio, will rebuild its power plant which was recently destroyed by fire. Henry Gebhart is superintendent.

Wellsville, Ohio, will build a municipal electric-light plant.

The Frankenmuth Light & Power Co., Frankenmuth, Mich., is considering the installation of an electric-light and power plant.

The city council, Mason City, Ill., has voted to install a new boiler in the municipal electric-light plant. Henry Hodshire is superintendent of plant.

The Public Service Co. of Northern Illinois is planning to rebuild its electric-light plant at Havana, Ill. J. M. Strasser, Streator, Ill., is superintendent.

WEST OF THE MISSISSIPPI

H. C. Hatch will construct an electric-light plant in George, Iowa.

Kellerton, Iowa, is considering the installation of an electric-light plant.

M. R. Faber and associates will install an electric-light plant at Remsen, Iowa.

Francis C. Cary, Minneapolis, Minn., will install an electric-light plant at South St. Paul, Minn.

St. Peter, Minn., is considering the installation of a street-lighting system.

D. W. Harlan has purchased the local electric-light and ice plants at Council Grove, Kan., and will expend about $10,000 on improvements.

Worley & Black, Reliance Building, Kansas City, Mo., are preparing plans for a water-works system and electric-light plant for Fowler, Kan. Estimated cost, $16,500. Grant Taylor is city clerk.

J. W. Lough & Co., Scott City, Kan., are preparing to install an electric-power plant, at a cost of $100,000.

Marquette, Neb., is considering the installation of an electric-light plant.

Esmond, N. D., is considering the installation of an electric-light plant.

Argenta, Ark., is considering the installation of a 500-hp. engine direct connected to a 300-kw. three-phase alternator and boilers, in the municipal electric-light plant. S. J. Chapman is superintendent of the plant.

Karnstadt, Wade & Stair will install an electric-light and power plant at Luling, Tex.

The Ordway Electric Light & Power Co., Ordway, Colo., will build an addition to its power plant. M. S. Slaughter is manager.

The Colorado Springs & Interurban Ry. Co., Colorado Springs, Colo., will install a 1250-kw. high-pressure turbine. C. E. Bibb, West Rio Grande St., Colorado Springs, is chief engineer of power station.

Henry Shaw, Las Vegas, N. M., will install an electric-light plant at Wagon Mound, N. M.

P. H. Patriarche and associates are considering the construction of a power plant on the Truckee River, near Reno, Nev.

The Fresno Consumers' Ice Co., Fresno, Calif., plans to increase the capacity of its plant.

Tropico, Calif., is considering the installation of a municipal electric-light plant.

The Union Ice Co., Winters, Calif., contemplates the erection of a new ice plant.

C. A. Smith, Pasadena, Calif., and Robt. A. Graham, Esperanza, Calif., are planning for the construction of a packing and cold-storage plant at Guaymas, Mex. Estimated cost, $50,000.

Fallon, Calif., is considering the installation of a municipal electric-light and power plant to cost about $15,000.

Palo Alto, Calif., is considering the installation of a 500-kw. turbo-generator in the municipal electric-light plant at a cost of $12,500. J. F. Byxbee, Jr., is city engineer.

Verdun, Que., is contemplating the installation of an electric-light plant and water-works system.

Bids will be received by G. R. Geary, chairman of board of control, Toronto, Ont., until noon, Mar. 26, for electrical equipment at the pumping station including one 200-hp. vertical motor, two 75-hp. vertical motors, one 9-panel switchboard.

Bids will be received by E. B. Bonnell, city clerk, Moose Jaw, Sask., until Mar. 18, for four centrifugal pumps, two motors, wiring, switchboard and accessories. Specifications are on file at the office of Walter J. Francis & Co., consulting engineers, Montreal, Que.

ENGINEERS WANTED

Under this head are published, without charge, advertisements of actual vacancies in power plants for operating engineers and assistant engineers. For the guidance of applicants for advertised positions, the advertisements must contain particulars to show the class of service demanded and the wages paid. Employers who desire to advertise under this head are requested to write for a blank form of application.

COMPETENT assistant engineer wanted for electric-light plant in central New York State; Heine boilers, 2000 hp.; 500-kw. Curtis turbines and Corliss engines; 2300-volt, three-phase alternators, operating in parallel; usual auxiliary equipment; engineer must be able to use various testing instruments and supervise repairs; thoroughly competent and qualified man required; wages, $85 per month, 10 hours' daily work, vacation in summer; chance to advance. Box 609, Power.

HELP WANTED

Advertisements under this head are inserted for 25 cents per line. About five words make a line.

SALESMAN — Thoroughly competent steam specialty salesman; one that can sell high-grade goods. Address "M. M. Co.," Power.

AGENTS for first-class steam specialty in use throughout United States. Address C. S. Wood, 410 S. 15th St., Philadelphia, Penn.

THE VULCAN SOOT CLEANER offers an exceptional opportunity for power specialty salesman Address G. L. Simonds & Co., 802 Steinway Bldg., Chicago, Ill.

SALESMAN for steam-heating system and specials; one acquainted in New England; salary and commission to the right one; excellent opportunity to right man. Address Box 613, Power.

ACTIVE AGENTS wanted to handle an engine room and water-works specialty of great merit; we will give exclusive territory; a well paying proposition for energetic man. Box 599, Power.

AGENTS WANTED to sell forced draft blowers and damper regulators; will give exclusive territory to right parties; commission on sales. Turbo-Blower Co., 30 Church St., New York.

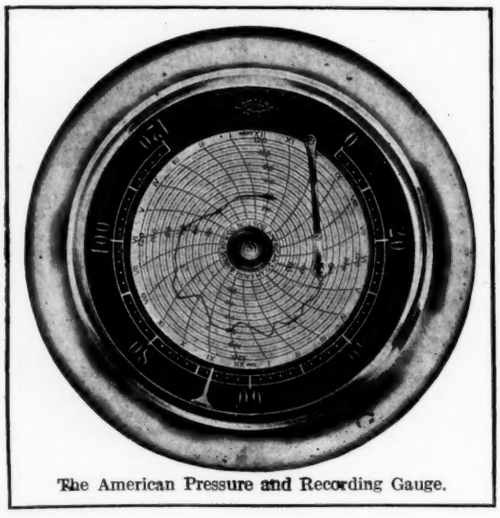
WANTED—Thoroughly competent experienced mechanical rubber salesman for the city of Chicago; must have an acquaintance among the large buying trade; state age and experience; also give references. Address Box 814, Trenton, N. J.

What The American Pressure And Recording Gauge Tells You

It tells you at all times what pressure was carried on the boiler at any time during its operation.

It tells you what is going on in the boiler room—keeps you in constant touch with things.

It tells you whether the fire is being carefully tended and shows when the fireman neglects his duty.



The American Pressure and Recording Gauge.

It can be installed in the manager's office or in any part of the plant.

For the manager it means a safeguard against careless firing.

For the engineer and fireman, if their work is ably done, it means a daily proof before the manager that they are the right men for their jobs.

It's a money saving device for small and large plants.

The workmanship of the device is "right." There are long years of experience back of it to make it so.

Write For Pocket Edition Catalog

Just sign and mail the coupon—

This Pocket Edition of the American Catalog is a handy book to have around for quick reference.

In it is listed its American line of money-saving devices. Every engineer and power plant man should have a copy.

Start the first step toward getting yours today by mailing the coupon.

SIGN AND MAIL COUPON

American Steam Gauge & Valve Mfg. Co.
Boston, Mass. 3-19-12

Send me your Pocket Edition Catalog

Name. . . . . .

Address. . . . . .

American Steam Gauge & Valve Mfg. Co.
Boston, Mass.

Sales Offices in New York, Chicago, San Francisco, Pittsburg, Atlanta, Montreal

THE BEST HOUSES EVERYWHERE SELL OUR PRODUCTS